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A Suspension System for a Military Tracked Vehicle was developed using a compressible silicone based fluid to provide both springing and dampening. The Suspension System will allow full retraction of the track system. The lightweight system provides seventeen inches Joiner, four inches Rebound and a 3.5 g Load Capacity for a Static Loading of 5100 pounds each.

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DEVELOPMENT OF A RETRACTABLE COMPRESSIBLE FLUID SUSPENSION SYSTEM

Prepared by
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TASK 1 TECHNICAL REPORT
FINAL DESIGN
Contract No. N00167-87-C-0111

Prepared for
David Taylor Research Center
Bethesda, Maryland 20084

1 June 1988

Approved:

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TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

LIST OF ILLUSTRATIONS (Continued)

LIST OF TABLES

I. INTRODUCTION

Southwest Research Institute (SwRI), under Contract No. N00167-87-C-0111, is developing a retractable compressible fluid suspension system for David Taylor Research Center (DTRC). This report summarizes the final design of this suspension system and is provided as required in DD Form 1423, CDRL No. A001. The second task of this project is to fabricate twelve units plus two spares for installation on the Propulsion System Demonstrator (PSD), a high waterspeed amphibious tracked vehicle, being developed for DTRC.

The objective of this project was to develop a lightweight, reliable, cost effective, retractable suspension system. The primary functional requirements for each unit of this system are listed below.

1. Provide a total of 21 inches of vertical roadwheel travel, of which 17 inches is jounce, and 4 inches is rebound from the normal static position.

2. The unit must have a design loading at the full 17-inch jounce position of 3.5 times its normal static loading.

3. The spring rate shall be designed to provide a vehicle natural heave frequency on the order of 1 Hz.

4. The unit must be capable of withstanding a worst-case loading of 100,000 lbs vertical as the roadwheel bottoms out on the sponson, and 20,000 lbs horizontal applied at the track.

5. The units shall have the capability to be adjusted for differing damping rates and static loads as required by each unit's location on the vehicle so that optimum total vehicle suspension performance can be achieved.

6. The units shall be capable of retracting to the full jounce position for the water propulsion mode.

7. The twelve units on the vehicle shall have a total weight of less than 3,800 lbs (317 each) including all components from the hull attachment point up to but excluding the roadwheels. This weight should also include all other suspension components (pumps, hoses, etc.).

8. The units shall have sufficient heat rejection capability and be relatively insensitive to ambient air temperature changes.

The design concept used to accomplish these requirements is unique in that it utilizes a compressible silicone fluid produced by Dow Corning Corporation as the compliant spring mechanism. This fluid serves also as the damping fluid. Liquid Spring Corporation will supply the struts for this system. They manufacture suspension struts which use this same concept for off-road mining trucks.

The design described herein utilizes the latest materials technology, including a composite high pressure vessel and high alloy aluminum structural materials.

This report will discuss this design in detail. The discussions will include the factors involved in defining the geometry, the parameters by which the bearings were selected, the stress analysis of the housing and roadarm using finite element techniques, and the selection of seals. Other discussions include an analysis of the damping characteristics of the units, the strut design, the extension and retraction controls design, a description of the composite pressure vessel, corrosion resistance measures, and the fabrication and test plan. A breadboard test unit was designed and fabricated to demonstrate the concept and to perform preliminary tests on some of the more critical areas of the design. The progress of testing of this unit is also discussed.

II. SUMMARY OF DESIGN

Figure 1 summarizes the fit, form, and function of the design. Figure 1a shows the basic dimensions of the unit and its degree of angular deflection capability. Full rebound to jounce capability as specified by DTRC is provided. Figure 1b shows the spring force versus deflection characteristics. Figure 1c shows the range of damping that will be provided by simple external adjustment. Finally, Figure 1d shows the projected weight breakdown of the design which totals 247.2 lbs per suspension unit for a total weight including other hardware of 3,352 lbs. This is 19.6 percent less than originally proposed and 11.8 percent less than DTRC's specified goal.

The system consists of a roadarm which is pivoted on the suspension housing as shown in Figure 1a. The roadarm extends past the roadwheel hub to provide a pivot location for the external linear actuator or strut. It is the rod of this strut that compresses the fluid during compression. Fluid is routed to the housing through the strut rod and strut rod end pivot shaft. A boot is attached to the rod end to provide external protection of the rod and to assist in cooling the oil by pumping air over the actuator. The strut is of the same robust design as that used on commercial off-road mining trucks.

Damping is provided when oil is pump d by the strut piston through an externally adjustable orifice located in the strut rod pivot shaft. A check valve will assure that damping is provided in jounce only. A relief valve located in the strut piston will provide the step in the damping curve to limit damping forces at high vertical roadwheel velocities. The strut will also serve to raise the roadwheel during retraction by bleeding the piston end and pressurizing the rod end.

Spring force for the suspension struts comes from contracting the strut which compresses the fluid. To provide adequate fluid volume so the correct spring rate can be achieved, additional fluid volume is contained in a composite pressure vessel located on the inside of the housing. The housing can efficiently provide support of external roadarm loads, strut loads, a location for valves, and space for the pressure vessel.

This design is well-suited for application on a Marine Corps amphibious vehicle because it maximizes the potential for reliability and maintainability. Reliability is enhanced over the conventional gas-oil systems because the gas-oil interface is eliminated. One fluid provides both spring and damping forces. Consequently, no gas leaks are possible.

Fail-safe operation is ensured when the engine and/or electrical system are not operational. Because the unit uses a compressible fluid, and because poppet-type valves are used, leakage should be minimized if not eliminated. Commercial off-road vehicles using Liquid Spring struts to support their entire weight demonstrated no leakage during months of operation and storage. Thus, the vehicle will not squat because of hydraulic leakage. Once charged, each unit acts as an independent passive suspension system. Even if damage occurs on one unit

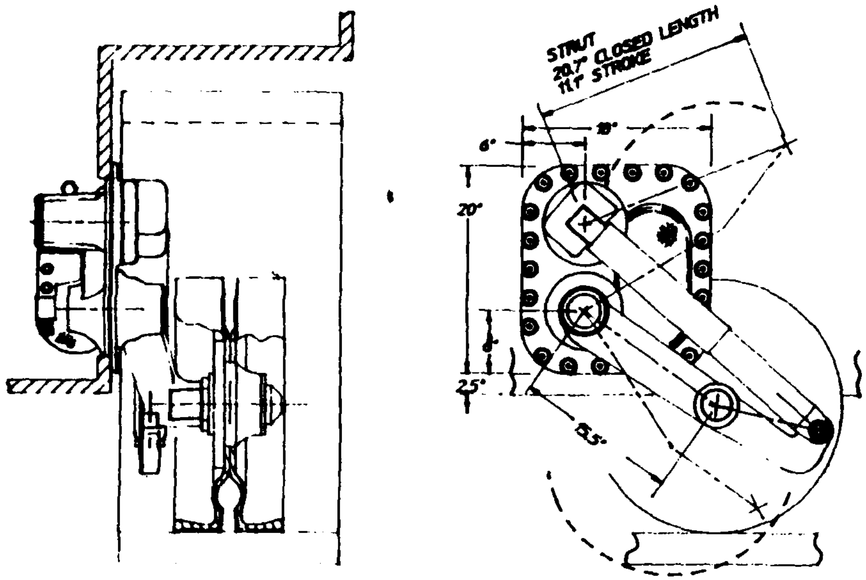


FIGURE 1A

RETRACTABLE SUSPENSION
STATIC PRESSURE AND FORCE-vs-DISPLACEMENT

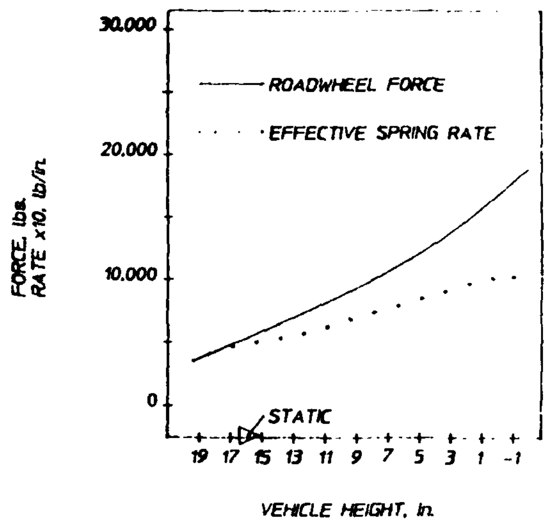
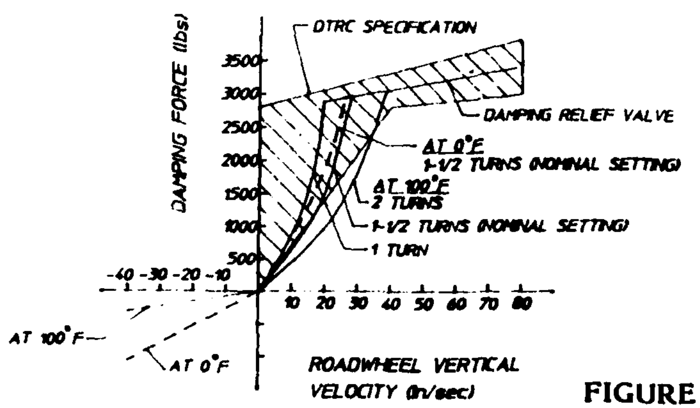


FIGURE 1B

DAMPING FORCE-vs-ROADWHEEL VELOCITY



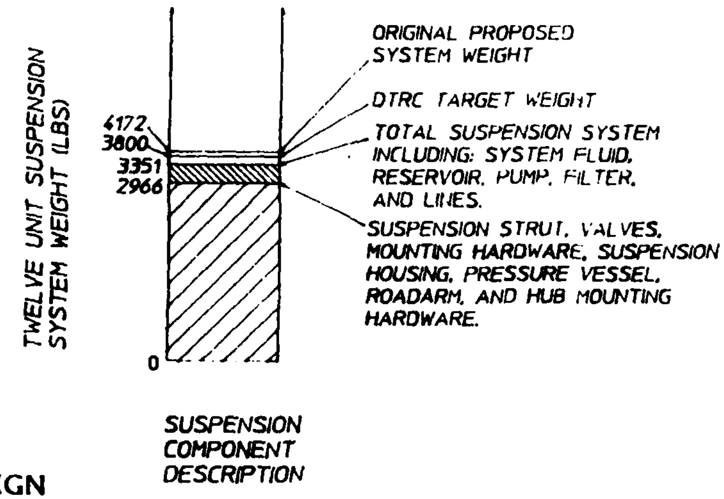


FIGURE 1. SUMMARY OF DESIGN

or the supply pump, each unit will continue to operate as dictated by its last adjustment setting.

The extension/retraction control system not only retracts the suspension units for the water propulsion mode and extends them for the land propulsion mode, but it continually monitors the system while extended to assure the proper setting independent of any changes in the fluid temperature. Poppet valves are used between the unit and its extension/retraction control valve. This practically assures zero fluid flow when they are not externally actuated. They are also less sensitive to contamination. The extension/retraction control valve is a spool-type directional valve that provides the proper amount of fluid in the unit. It senses the suspension unit's position and pressure independent of a particular unit's position or applied force and will adjust accordingly.

To retract the suspension unit a separate pilot pressure source is activated. This overrides the normal pressure feedback function, positioning the valve to depressurize the unit and subsequently pressurize the rod side of the strut to pull the roadarm fully up.

To interface the suspension system to the vehicle, a power source (pump) is required to provide pressurized flow to each unit, as well as a reservoir and appropriate valving and lines. Three lines are required to each unit: supply pressure, return to tank, and pilot pressure for activating the retract mode.

In the interest of minimizing the weight, the suspension units were not designed for ballistics protection. In particular, the boot on the strut is only 0.125-inch thick steel, and some areas of the housing are only 0.375 inch thick.

III. DETAILED DESIGN PRESENTATION

The design of this suspension system includes many design challenges. These design challenges and how they were addressed will be discussed in this section. A cross-sectional view of the layout of the suspension system is shown in Figure 2.

A. Suspension System Geometry

To determine the geometry of the suspension system to obtain optimum performance it was first necessary to define the equations that describe the action of the system and the controlling equations. It was also necessary to define "figures of merit" from which to judge the optimum design. A computer program was written to analyze this suspension system and perform an iterative procedure to search for the optimum design.

The effective spring characteristics of the suspension unit at the roadwheel is dependent upon the geometry of the roadarm and strut pivot locations. It is also dependent upon the rod diameter of the strut and the volume of fluid being compressed.

The compressibility relationship of the Dow Corning 200, silicone/fluid is as follows. General information on its other properties is included in Appendix A.

$$P = (K)(VR)^B \tag{1}$$

where: P = **fluid pressure, psi**
K = **fluid constant, 422,936**
VR = **volume ratio = displaced volume/free fluid volume**
B = **fluid exponent, 1.3**

The spring action takes place as the strut rod is displaced into the strut cylinder, displacing fluid volume. As fluid volume is displaced, the volume ratio, VR, increases and so does the pressure in the strut. Since the pressure in the strut acts on the area of the rod to force it back out, the reaction force increases also. Therefore, as the rod is displaced the reaction force increases much like a spring. Since the compressibility equation of the fluid is an exponential function, the spring rate of the strut is nonlinear, increasing with compression of the strut.

The compressible fluid strut also incorporates a damping function such as a shock absorber. This is accomplished by making the bore of the cylinder larger than the rod and adding a piston to the end of the rod. Thus as the rod displaces in and out, fluid will transfer from one side of the piston to the other through a restriction resulting in a damping force that varies with the velocity of the rod motion.

When the characteristics of the strut are combined with the kinematic relationships of the roadarm, designing the system for optimum performance becomes more complex. A computer program was written to analyze the system and to search for the optimum geometry. A listing of the program and the input data file are included in Appendix B.

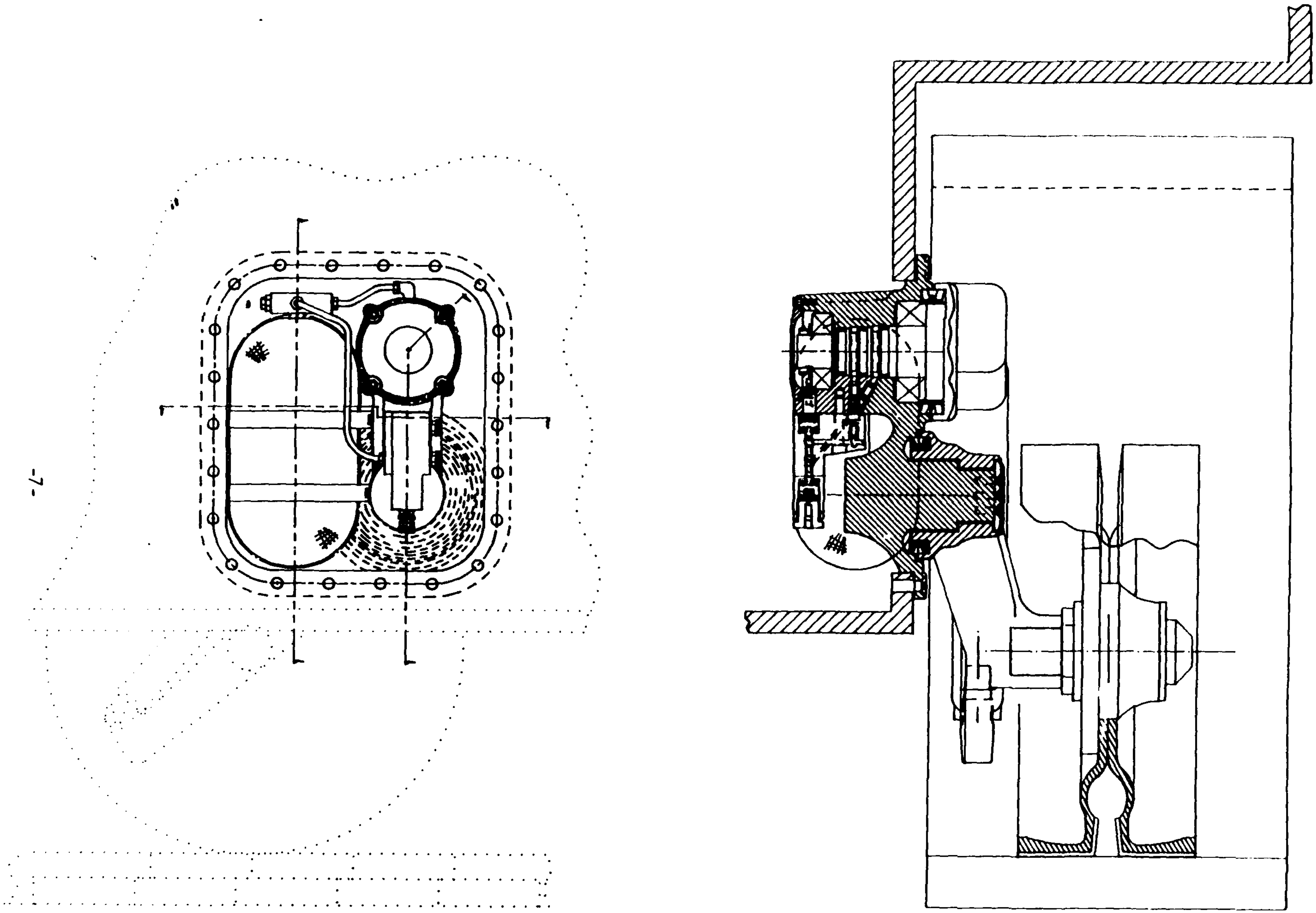


FIGURE 2. SUSPENSION UNIT LAYOUT

Six variables were adjusted in the model, all of which are geometric positions of the suspension system pivot points relative to the vehicle and relative to one another. These parameters are illustrated in Appendix B along with the resulting optimum values.

Several geometric boundaries had to be placed in the model such as limiting how close the strut could come to the roadarm pivot in the full rebound position as well as placing a lower limit on the tare length of the strut (the fully compressed length minus the stroke).

Some of the parameters that were used as "figures of merit" from which to measure how optimum a design is include the minimization of the free volume of fluid (minimize weight), maintaining a spring rate at the normal static position close to or above 528 lbs/in. (the rate required for a 1 Hz natural frequency for one unit's share of the 62,000-lb vehicle weight), minimize the free volume to maximum displaced volume ratio (minimizes temperature effects on position), and maximizing the minimum change in spring rate (insures that the spring rate continues to rise with deflection). Each of these parameters were weighted to reflect their importance and effect on the overall figure of merit.

The program begins with a starting set of variables and determines the figure of merit by simulating the stroking of the suspension unit from full rebound to full jounce. Then every combination of any two of the variables is varied up or down a small incremental distance and the variables producing the greatest positive rate of change in the figure of merit are updated to the better value. The process is repeated until no more improvement is made.

The optimized geometry produced spring characteristics as shown on Figure 3. This plot displays the strut pressure, the strut force, the roadwheel force, and the effective spring rate at the roadwheel versus the vehicle's height or ground clearance where 16 inches is the normal static position. This plot also indicates a steadily increasing spring rate which is two times higher at full jounce than at the static position. The roadwheel force is 3.5 times higher at full jounce than at static. Table 1 is a listing of the values used in this plot.

The actual spring rate achieved with this design at the static position was 504 lbs/in., which results in a natural heave frequency of 0.997 Hz for a 62,000-lb vehicle with twelve units.

The strut rod is 1.5 inches in diameter and the cylinder bore is 2 inches in diameter. At full jounce the pressure reaches 18,000 psi. At full rebound the pressure is 3,660 psi (precharge). The total amount of free volume of fluid required in each unit is 314 cubic inches. The fluid free volume is compressed to a physical volume of 306 cubic inches resulting in the 3,660 psi precharge pressure at full rebound. Of this 306 cubic inches of physical volume, 35.5 cubic inches is provided by the strut, 6 cubic inches by the communication passages, and 264.5 cubic inches by the pressure vessel.

The total amount of travel that the suspension system is designed for is 21 inches, 17 inches jounce and 4 inches rebound. At full jounce the roadwheel

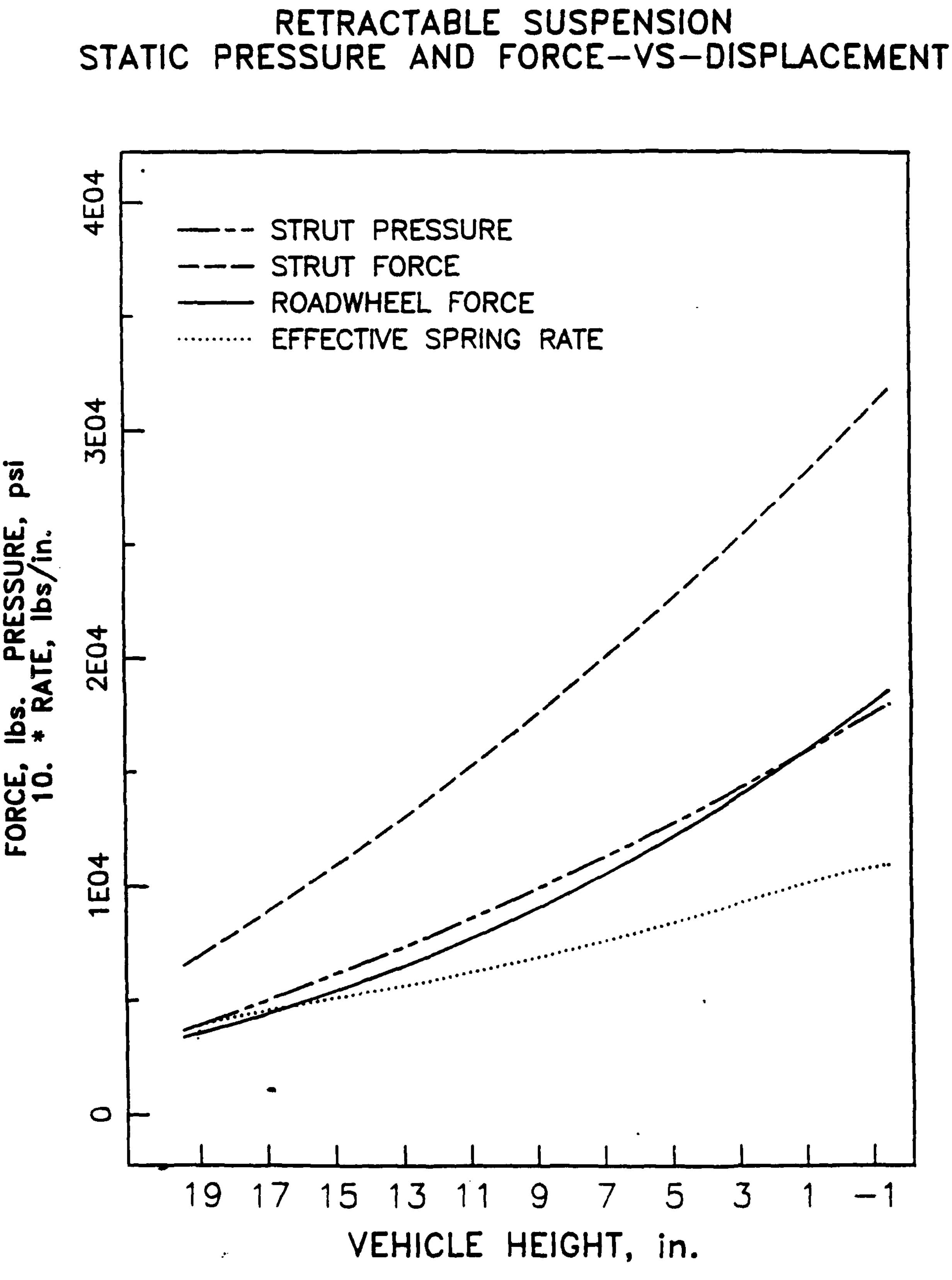


FIGURE 3. SPRING CHARACTERISTICS

Table 1. Spring Characteristics

Y	F(HUB)	F(STRUT)	P(STRUT)	SPRING RATE	STRUT ANGLE
19.5	3365.8	6468.8	3660.9	0.0	-58.3
19.0	3562.5	6944.9	3930.3	393.4	-55.8
18.5	3769.3	7422.6	4200.7	413.8	-53.4
18.0	3984.7	7903.5	4472.9	430.8	-51.2
17.5	4207.6	8388.3	4747.2	445.8	-49.1
17.0	4437.5	8877.5	5024.0	459.8	-47.0
16.5	4674.1	9371.6	5303.7	473.2	-45.0
16.0	4917.3	9871.0	5586.3	486.4	-43.1
15.5	5167.0	10375.8	5872.0	499.3	-41.2
15.0	5423.2	10886.3	6160.9	512.4	-39.4
14.5	5686.1	11402.9	6453.3	525.8	-37.6
14.0	5955.6	11925.6	6749.0	539.1	-35.8
13.5	6232.0	12454.4	7048.3	552.7	-34.1
13.0	6515.3	12989.7	7351.3	566.6	-32.4
12.5	6805.7	13531.5	7657.9	580.8	-30.7
12.0	7103.4	14080.1	7968.4	595.5	-29.0
11.5	7408.6	14635.4	8282.6	610.3	-27.3
11.0	7721.4	15197.5	8600.7	625.6	-25.6
10.5	8042.1	15766.7	8922.9	641.4	-23.9
10.0	8370.8	16343.0	9249.0	657.4	-22.2
9.5	8707.8	16926.5	9579.2	674.0	-20.6
9.0	9053.2	17517.4	9913.6	691.0	-18.9
8.5	9407.5	18115.7	10252.2	708.4	-17.2
8.0	9770.6	18721.5	10595.1	726.3	-15.5
7.5	10142.9	19335.1	10942.3	744.7	-13.8
7.0	10524.7	19956.4	11294.0	763.5	-12.1
6.5	10916.1	20585.6	11650.1	782.8	-10.4
6.0	11317.4	21222.8	12010.7	802.6	-8.6
5.5	11728.8	21868.1	12375.9	822.8	-6.8
5.0	12150.5	22521.8	12745.8	843.5	-5.0
4.5	12582.8	23183.7	13120.4	864.5	-3.2
4.0	13025.7	23854.0	13499.7	885.8	-1.4
3.5	13479.4	24532.9	13883.9	907.4	.5
3.0	13944.0	25220.4	14273.0	929.3	2.4
2.5	14419.6	25916.6	14667.0	951.1	4.3
2.0	14906.1	26621.7	15066.1	973.0	6.3
1.5	15403.3	27335.6	15470.1	994.4	8.4
1.0	15911.1	28058.5	15879.2	1015.5	10.4
.5	16428.9	28790.2	16293.3	1035.6	12.6
0.0	16956.0	29530.9	16712.5	1054.3	14.8
-.5	17491.7	30280.6	17136.7	1071.3	17.1
-1.0	18034.4	31039.0	17565.9	1085.4	19.4
-1.5	18582.4	31806.0	18000.0	1096.0	21.9

bottoms out on the upper section of the track and the vehicle hull sponson. Additional loads beyond what the suspension unit supports at full jounce will be carried by the sponson, up to a total of 100,000 lbs as specified in the RFP. Because of deflection and wear in the tracks, the roadwheel may travel an additional 0.5 inch vertically after contact with the sponson is made. The suspension unit is installed on the vehicle so that the nominal travel is reduced from 21 inches to 20.5 inches to allow for this deflection.

B. Bearing Analysis

There are six flexible joints in this suspension system, each requiring a bearing. These joints are the roadwheel hub, the roadarm pivot, the attachment point of the strut to the roadarm, the strut pivot shaft, the strut-to-strut pivot shaft trunnion, and the strut rod and piston sliding interface. All of these will be reviewed here except for the last item which will be discussed further in the section on seals since its load is induced by seal friction.

1. Roadwheel Bearings

To minimize the design and fabrication cost of the suspension units, the roadwheel hub and spindle currently used on the AAV-7A1 amphibious vehicle are to be used on this suspension system. The bearings used are the M5519081-105 and the MS519081-86 tapered roller bearings rated for 7000 and 5000 lbs radial load, respectively (at an L10 life of 3000 hours at 500 rpm). With a maximum vehicle speed of 45 mph and a roadwheel diameter of 24 inches, the roadwheels will turn at a maximum speed of 630 rpm. The speed adjustment factor would then be 0.933.

The load duty cycle was assumed to be the same as used by TACOM on their seal test which is:

Table 2. Bearing Load Data Cycle

	Time	Max Load
40 cycles at 1/2 inch	8 sec	5,423 lbs
9 cycles at 3 inch	1.8 sec	6,805 lbs
1 cycles at 17 inch	.18 sec	18,400 lbs
Total	10.0 sec	

The adjusted weighted load is 7,014 lbs assuming the load is continuous at each of the maximum load values. Dividing by the speed factor, the design load should be 7,517 lbs. The combined load rating of the two bearings is 12,000 lbs, which is well above the design load.

2. **Roadarm Pivot Bearings**

The roadarm pivot bearings specified are DU self-lubricating bearings from Garlock Bearing Inc. There are two radial bearings, 3 inches and 4 inches in diameter, and each are 2 inches wide for a total projected area of 14 square inches. There are also two thrust bearings each with an area of 5.5 square inches. These bearings are steel backed with a bronze innerstructure and a PTFE-lead overlay. They are capable of taking a maximum unit load of 40,000 psi, steady unit load in one direction of 20,000 psi and an oscillating unit load of 8,000 psi.

The reaction load of the roadarm at full jounce is 55,000 lbs, which results in a unit load of 3,928 psi. The applied vertical load is 7.7 inches eccentric from the bearing center, thus applying at full jounce (18,400 lbs vertical) a twisting moment of 142,000 in-lbs. Assuming an average spread of the two bearings of 2.5 inches, their component reaction forces to this moment is 56,700 lbs each. The 3 inch bearing would have a combined unit load of 10,229 psi. With a 20,000-lb side load applied in on the roadwheel and an effective distance of 15 inches, the reaction unit load on the 3-inch bearing could be an additional 20,000 psi. Therefore the maximum unit load in the absolute worst case would be on the order of 30,000 psi, which is below the maximum limit.

To determine the loading at a steady operating condition, the static load case, the normal total reaction force is about 9,000 lbs. The effect of the eccentric loading adds a load component of about 16,000 lbs to each bearing. The resultant load for each bearing will be about 16,600 lbs or a unit loading for the 3-inch bearing of 2,770 psi, well below the 8,000 psi unit load limit for steady oscillating operation.

With a 20,000-lb side load the thrust bearings would have a unit loading of 3,600 psi, well within the design limits even if the load is not evenly distributed.

3. **Strut-to-Roadarm Bearing**

The joint where the strut connects to the roadarm must be able to allow misalignment and deflections. A spherical plain radial bearing was specified, such as the Torrington SF series with sealing elements. The specified 1.25-inch bearing has a dynamic load rating of 16,800 lbs and a maximum load rating of 52,000 lbs. With the strut force at 10,400 and 31,800 lbs at the static and full jounce positions respectively, this bearing should be more than adequate. An alternate filament-wound bearing will be evaluated on the breadboard test unit.

4. **Strut Pivot Shaft**

The pivot shaft for the strut has to carry the reaction force of the strut as well as maintain good alignment of the shaft in its bore for the close clearances required of the seals. Tapered roller bearings are best suited for this requirement. The radial loads on the outer bearing at the static position is 14,600 lbs and 43,800 lbs in the full jounce position. For the inner bearing the radial loads are 4,000 and 11,990 lbs respectively. When adjusted for the low speed and the duty cycle expected, the dynamic radial load rating for the outer and inner bearings is 4,990 and 1,380 lbs respectively. However, the maximum static load on a bearing should be no more than four times the basic dynamic load rating. Dividing the maximum radial loads of the outer and inner bearings would yield a required load rating of 10,950 lbs and 2,997 lbs, respectively. The inner bearing must also carry a high thrust load from the fluid pressure acting on the stepped diameters of the sealing interface. At 18,000 psi this thrust load would be 20,300 lbs. Dividing this by four would require a basic dynamic thrust load rating of 5,080 lbs. The specified bearings and their respective Timken part numbers and radial and thrust load ratings are listed below.

Table 3. Strut Pivot Shaft Bearing Capacity

	Timken Part No.		Bearing Load Rating, Lbs	
	Cone	Cup	Radial	Thrust
Outer	748-s	742	15,400	8,580
Inner	55200C	55443	6,190	9,390

These bearings are well within the design requirements described.

C. Stress Analysis

The primary items of concern in regard to strength are the housing and the roadarm. Both of these parts carry very high loads so they must have sufficient mass to provide adequate strength. Since the weight of these units is of great concern, it is imperative to optimize the use of material in these parts. To accomplish this, finite element analyses were performed on these parts.

1. Housing Stress Analysis

The analysis of the housing included two separate models. One was a three-dimensional solid model of the section of the housing that supports the roadarm pivot. The second model was of the entire housing but it utilized three-dimensional shell elements with numerically simulated thicknesses. The load condition simulated was the worst-case condition of full jounce and a 20,000-lb load forcing the roadwheel out. The finite element analysis software was the ANSYS/PC program from Swanson Analysis Systems, Inc.

Aluminum material was assumed in these analyses. The housing and the roadarm can be forged using a high alloy aluminum such as 7150 or 7175, which have yield strengths on the order of 76,000 psi and fatigue strength of about 23,000 psi.

The solid model of the housing section predicted a stress problem in the area where the roadarm pivot shaft joins the main body of the housing. By thickening up the material in that area and contouring the corner to relieve the stress concentration the stresses were reduced to a level of about 40,000 psi in this region. The area of highest stress was now in the area of the pivot shaft just above the place it joins the main body of the housing. The stresses in this area are on the order of 73,000 psi. The results of this model are shown in Figure 4 which is an isometric view of the model, and in Figure 5 which shows the stresses through a cross-section of the part. This model predicted some very high erroneous stresses at the end of the pivot shaft due to a crude means of applying the thrust loads to the shaft. This area was modeled more accurately as described in the following discussion.

A revised version of this model looked more closely at the stresses within the shaft and how the loads from the roadarm were applied. The previous model applied the load using simple beam elements connected directly to the solid elements. This revised version modeled the pivot end of the roadarm as a shell model and linked the roadarm model to the shaft with gap elements to simulate the gap that naturally occurs in a plain bearing interface. Figure 6 illustrate the roadarm model constructed of shell and beam elements. To limit the size of this model, the shaft was cut off and fixed at the point where it connects to the main housing body. This model showed that the stresses at the end of the shaft are not a problem and that the highest stresses are still near the base of the shaft. See Figure 7.

Two other load cases of concern are at minimum vertical load with 20,000 lbs force outboard and the normal static loading. The first is of concern since the moment arm of the applied side load is the longest of any position. The maximum stress for this condition is 59,000 psi as shown in Figure 8. The normal static case is of concern since these loads would be seen repeatedly and the housing may be subject to fatigue failure. The maximum stresses predicted in this case are 15,400 psi, which is below the endurance strength of the material (see Figure 9).

The shell model of the entire housing provided direction on sizing the thicknesses of the rest of the areas of the housing. It also provided an indication of what loads would be on the mounting bolts. Figure 10 shows the different element thicknesses used. Figure 11 shows the resulting stresses at the top level of the shell element and Figure 12 shows the stresses at the bottom level of the shell element. The maximum stresses predicted were in the same area and of the same magnitude, 73,000 psi, as predicted by the solid model.. This model also shows that the stresses are no more than 31,000 psi in other areas not modeled by the solid model, even in the thinnest, sections that are 0.375 inches thick.

The reaction forces at the nodes were also studied. The largest reaction force was at the constrained edge node directly below the pivot shaft. It reached a level of 47,000 lbs in tension. This would exceed the strength of a single 5/8-inch countersunk bolt. However, the actual housing design has two bolts that are located on each side of the nodal location. Combined, they will have sufficient strength to carry the load.

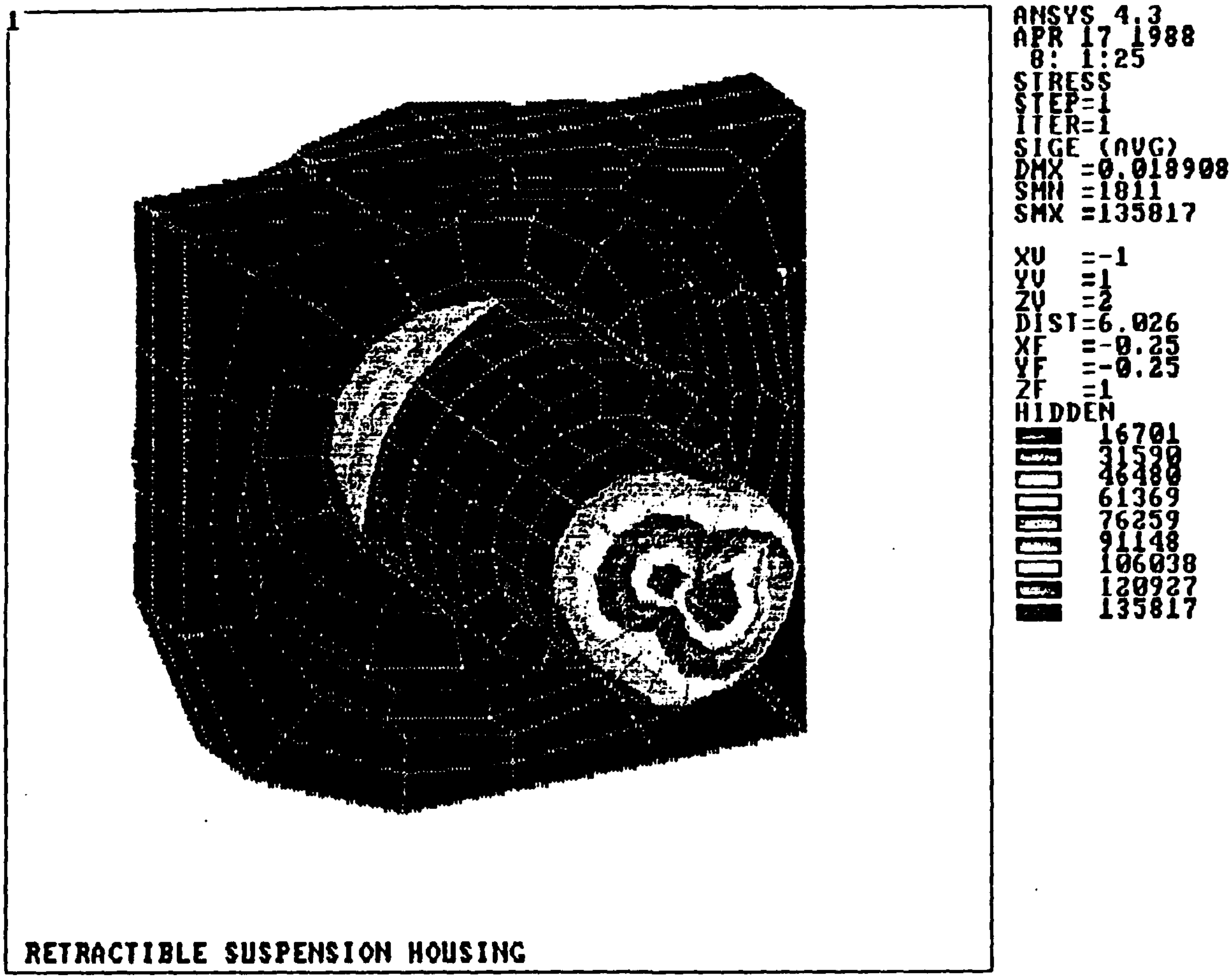


FIGURE 4. FEM HOUSING STRESS ANALYSIS, SOLID MODEL, ISOMETRIC VIEW

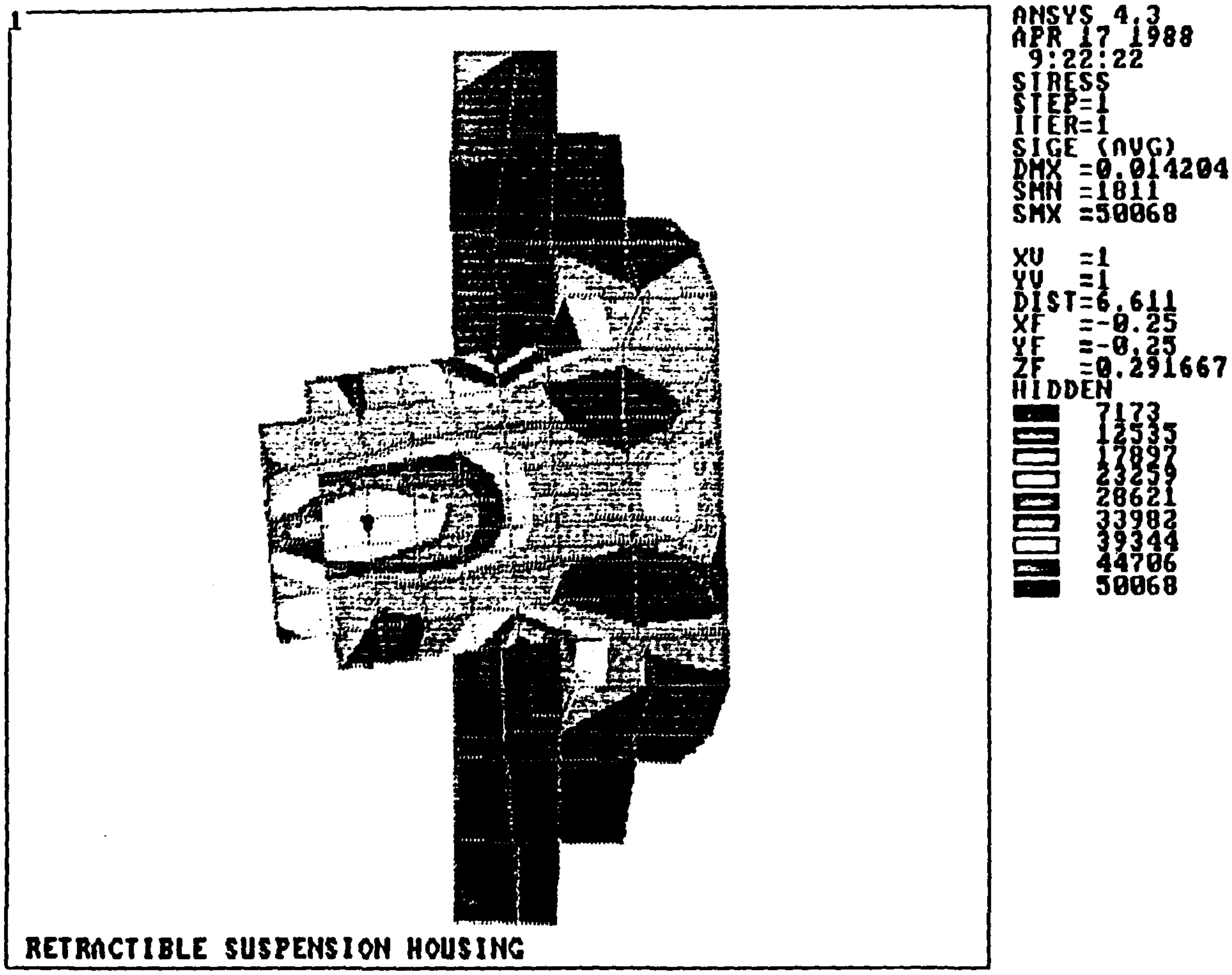


FIGURE 5. FEM HOUSING STRESS ANALYSIS, SOLID MODEL, SECTION VIEW

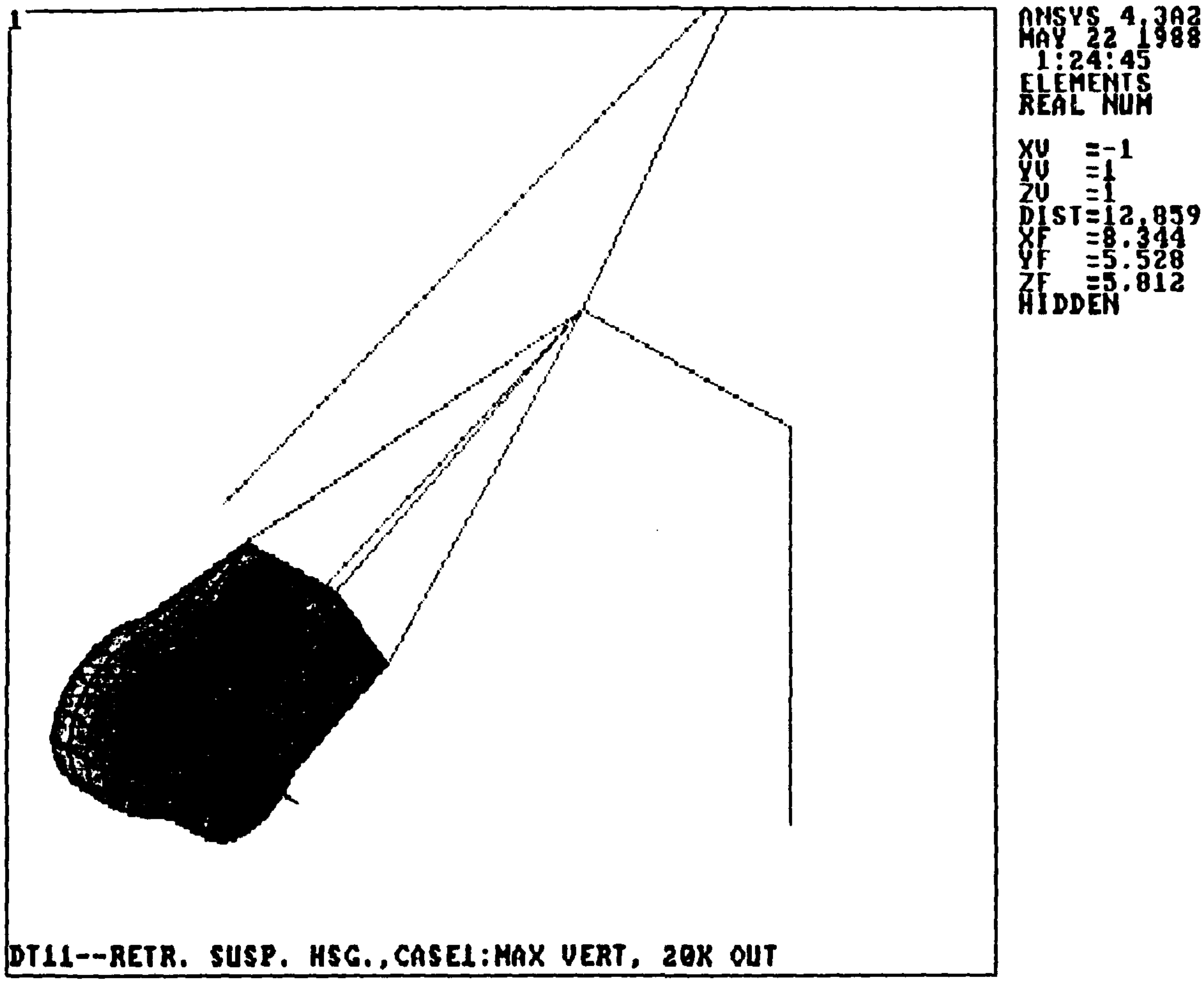


FIGURE 6. FEM ROADARM SHELL MODEL FOR APPLYING LOADS TO PIVOT SHAFT

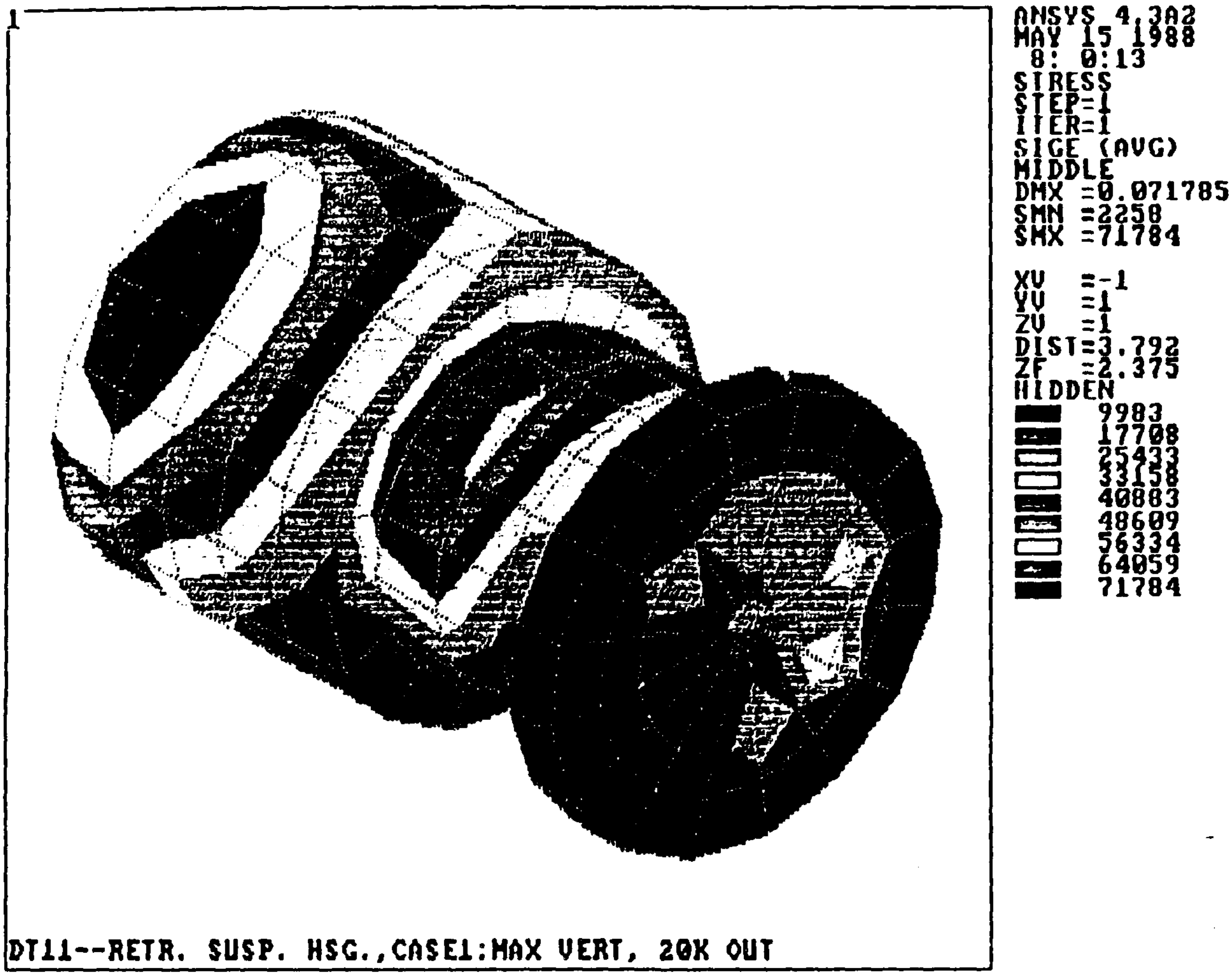


FIGURE 7. PIVOT SHAFT, SOLID MODEL

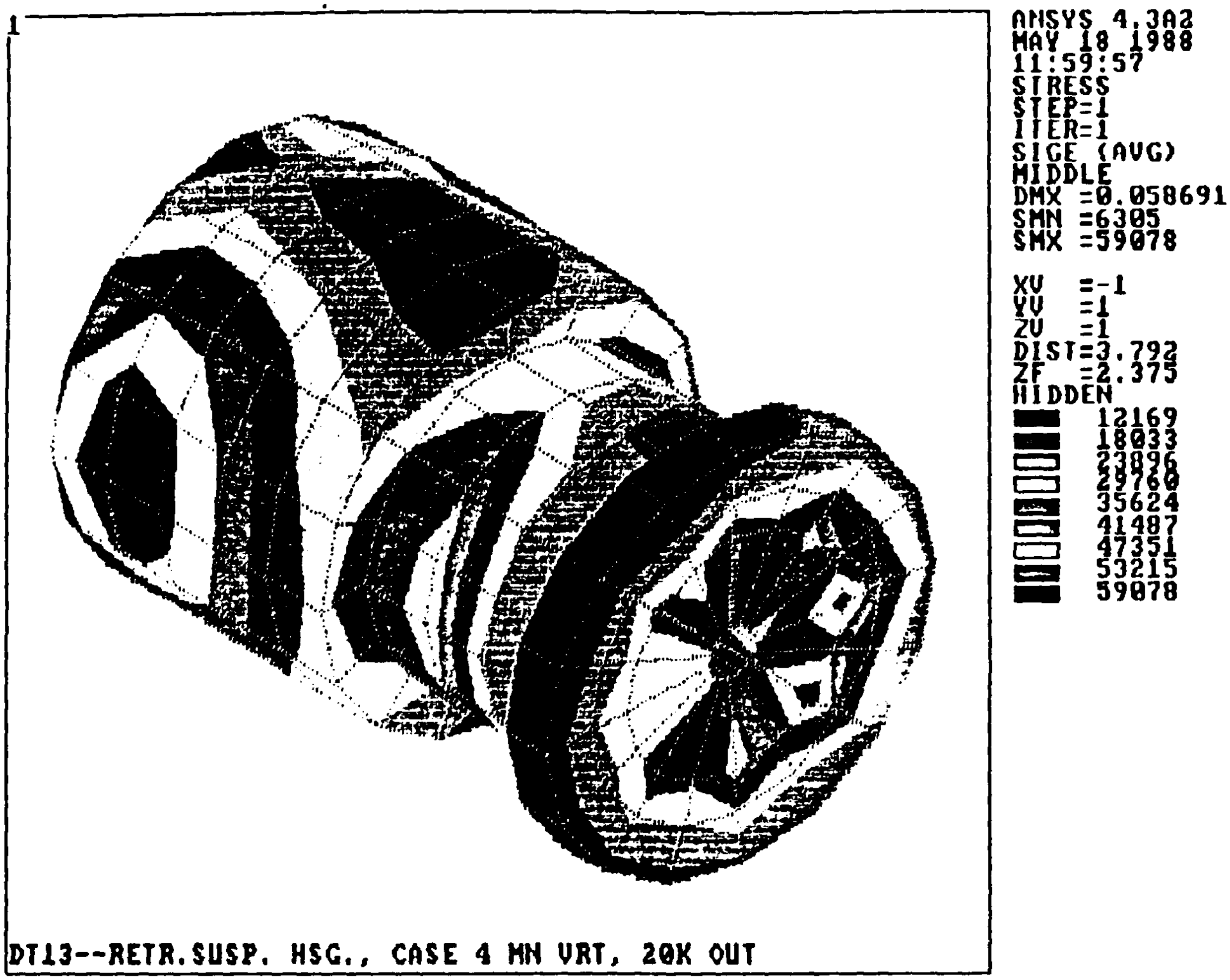


FIGURE 8. PIVOT SHAFT, SOLID MODEL AT MINIMUM VERTICAL POSITION AND 20,000 LBS OUTWARD LOAD

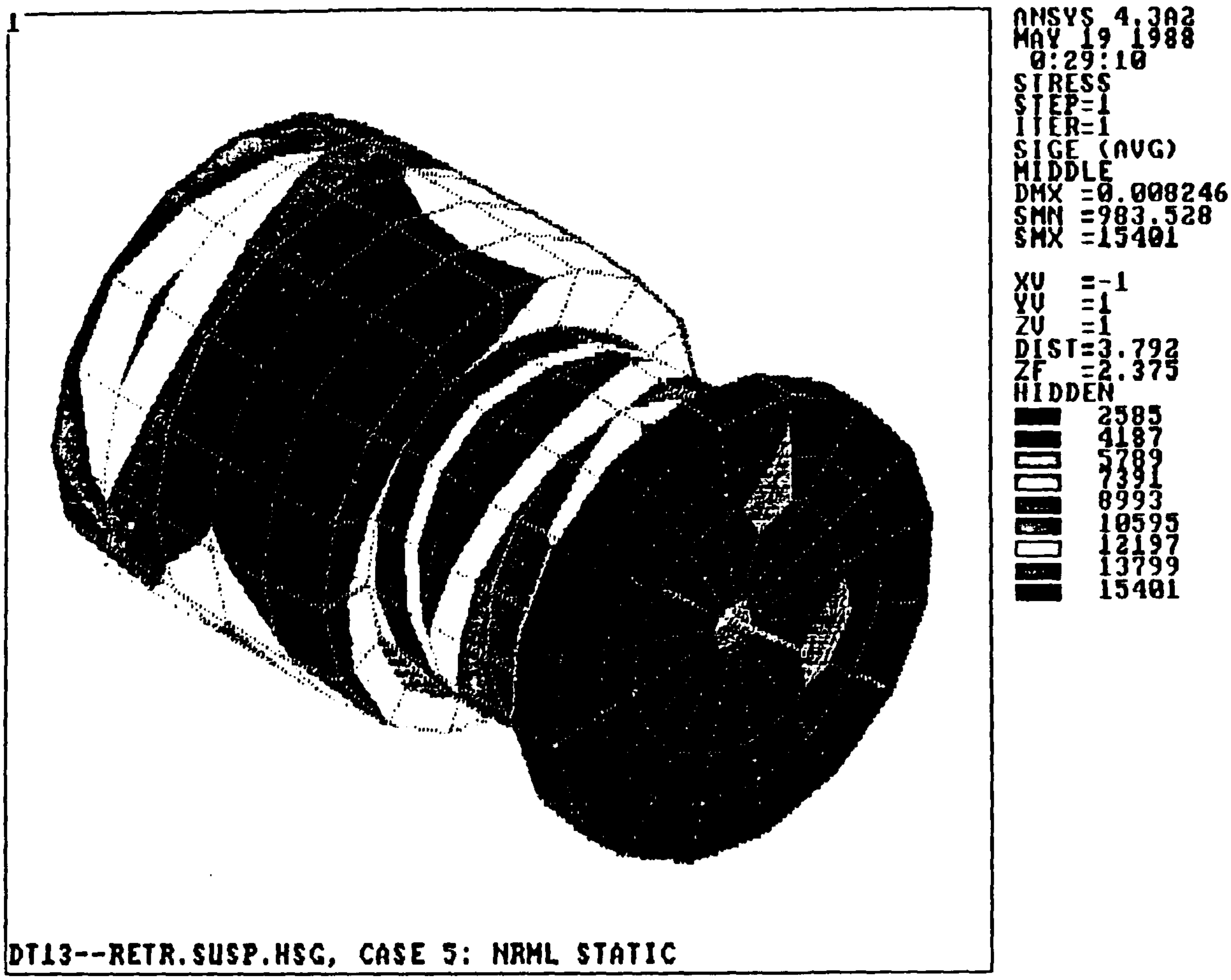
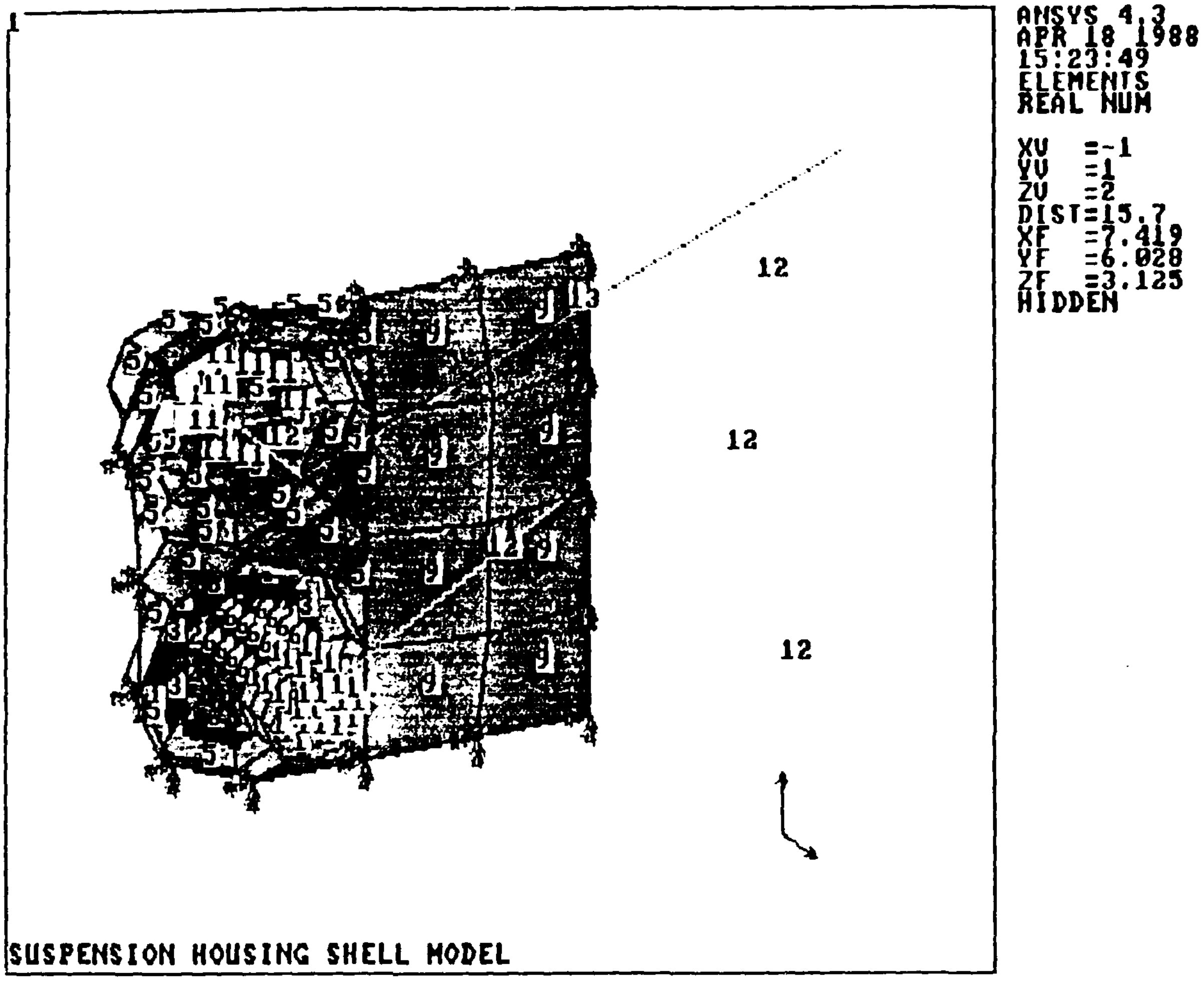


FIGURE 9. FEM PIVOT SHAFT SOLID MODEL AT NORMAL STATIC POSITION



Constant Number	Shell Thicknesses (inch)
1	1
2	1.5
3	1.5 to 1
5	0.5
6	1.5
9	0.375

FIGURE 10. HOUSING SHELL MODEL ELEMENT THICKNESSES

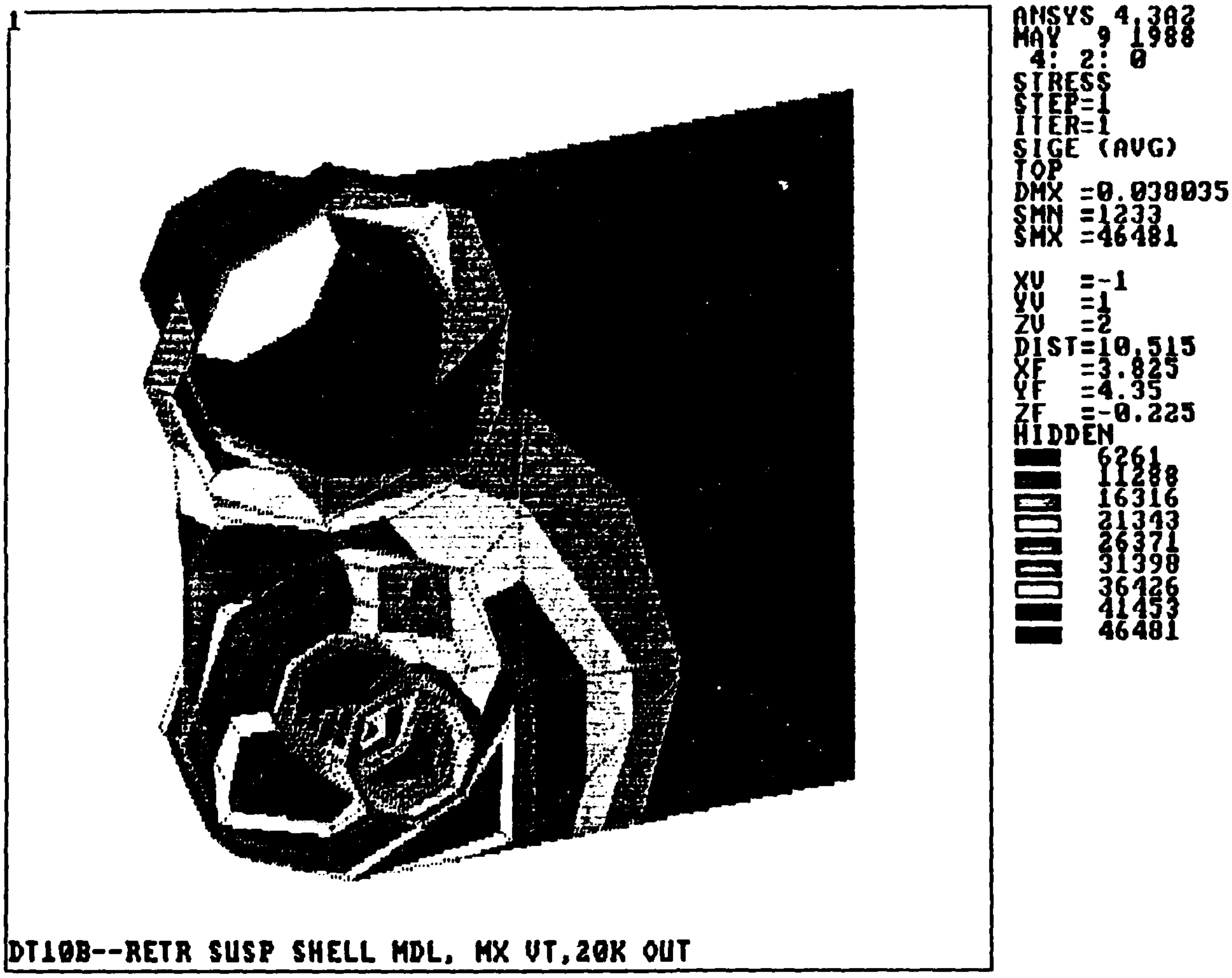


FIGURE 11. FEM HOUSING SHELL MODEL TOP LEVEL STRESSES

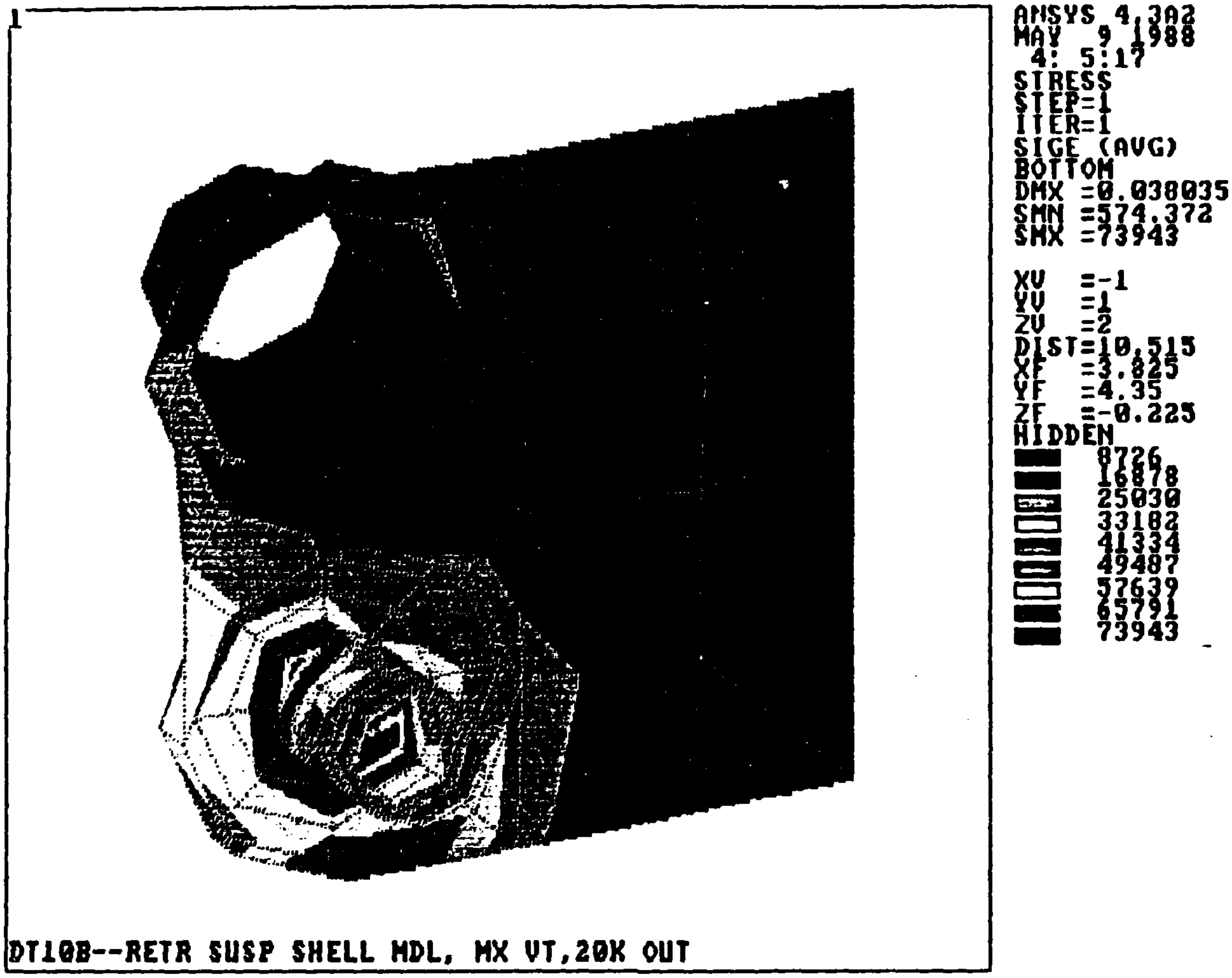


FIGURE 12. FEM HOUSING SHELL MODEL BOTTOM LEVEL STRESSES

2. Roadarm Analysis

The stress analysis using the finite element methods is incomplete at this time. Initial analyses assuming the use of steel predicted maximum stresses on the order of 120,000 psi. DTRC requested the use of lighter weight aluminum. In order to modify the model for using aluminum, more material had to be added as well as more elements to obtain sufficient accuracy. The results of this analysis are not yet available but will be provided at a later date when available.

3. Deflection Analysis

From the stress analyses previously described, deflections were also analyzed. From the housing shell model it was predicted that in the load case with maximum vertical load and 20,000 lbs horizontal outside load the roadarm pivot shaft would deflect 1.05 degrees. Projected out to where the load is applied at the roadwheel, the horizontal deflection would be 0.25 inch out. In the case of 20,000-lb horizontal load being applied inboard, the roadarm pivot shaft would deflect about 1.22 degrees. Projected to the same loading point, the horizontal deflection would be 0.29 inch inboard. About 75 percent of this deflection is in the housing itself with the rest being in the shaft. This does not account for any deflection of the roadarm or the roadwheel.

D. Fluid Seals

There are several interfaces requiring high pressure sealing elements, and several other interfaces requiring low pressure, dust seals. The high pressure seals are in the strut pivot shaft and the trunnion joint between the strut and the strut pivot shaft. The high pressure seals in the strut will be discussed in the section describing the strut. The low pressure seals are those that seal the roadarm bearings, the strut pivot bearings, and the strut trunnion bearings from the elements and seal in the fluid to lubricate the bearings.

1. Strut Pivot Shaft Seals

The most critical sealing interface is in the strut pivot shaft which must seal up to 18,000 psi of pressure and be free to move in an oscillating rotary motion. Two of these seals must prevent fluid from leaking into the tapered roller bearing cavities while under pressure of up to 18,000 psi. A third seal located between these two must contain the differential pressure (up to 4000 psi) induced by damping and by the retraction mode (up to 7500 psi).

The seals selected for this interface are provided by Green Tweed & Co., Ltd. The seal recommended for this application is the Ener-Cap TM seal with backup rings on each side (see Figure 13). This seal consists of an elastomer seal with a Teflon cap. The Teflon cap serves as a low friction sealing interface. The backup rings are made of moly-loaded nylon to serve as low friction, low wear, interfaces to prevent extrusion through the shaft clearances.

Since the strut pivot shaft seals will be under extremely high pressures, they will have a considerable amount of friction even though they

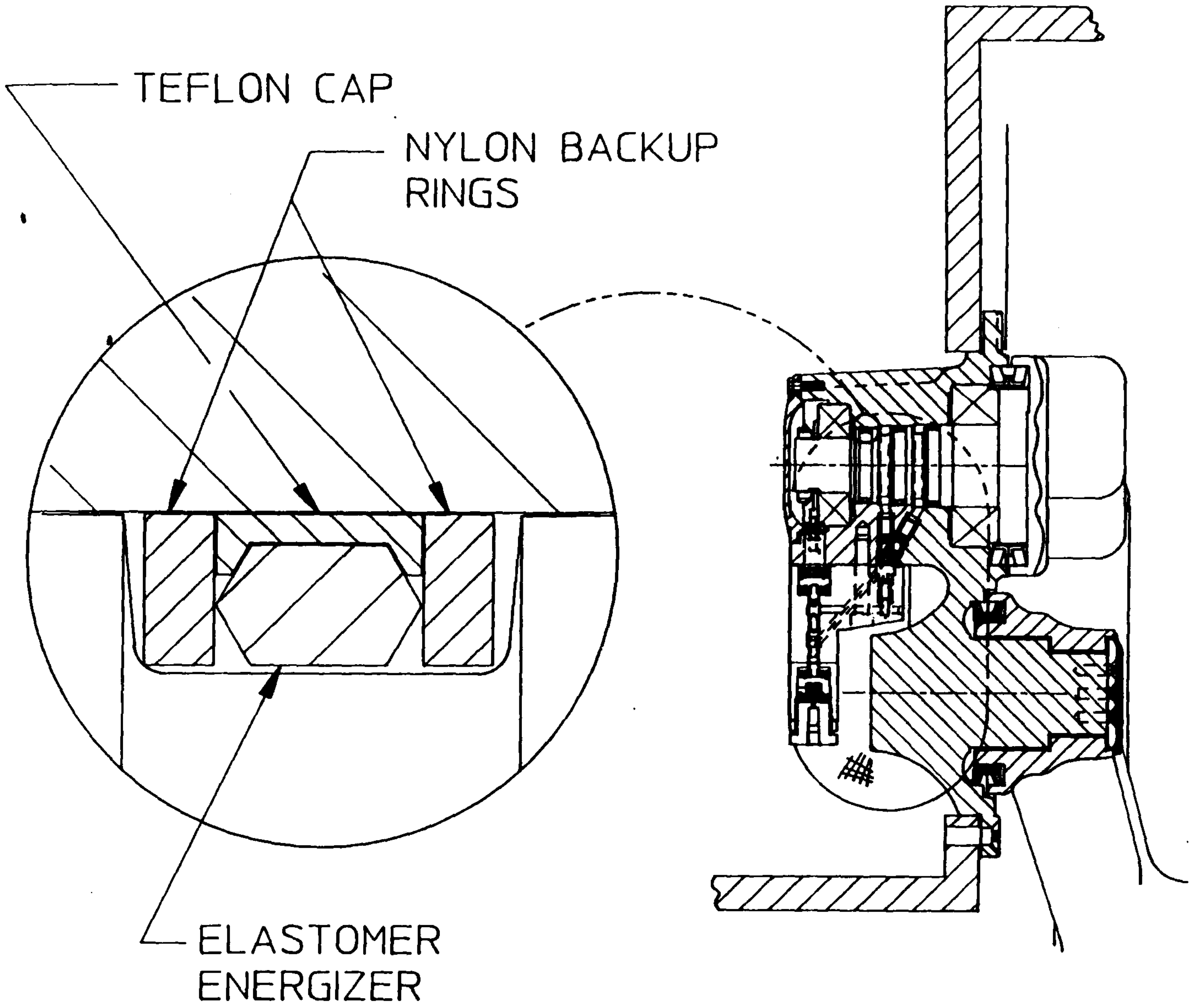


FIGURE 13. STRUT PIVOT SHAFT SEAL

contain Teflon materials. A simplified analysis of the resulting loads will suffice to reveal potential problems. If it is assumed that pressure applies a normal force to the seal cap equal to the pressure times the contact area of the cap and if it is also assumed that the coefficient of friction is 0.1 and that two seals are under pressure, a friction torque can be estimated. Based upon these assumptions, the torque (in-lbs) required to rotate the strut pivot shaft would be 0.95 times the pressure (psi). Thus, at 18,000 psi, the torque would be 17,100 in-lbs.

Since the strut has to rotate the strut pivot shaft, this load must be carried through the rod, the rod bearing, and the piston bearing. The worst case for the rod bearing will be at full rebound where it will have to carry about 535 lbs, or a unit loaded of about 316 psi. The worst case for the piston bearing is at full jounce where it will have to carry about 953 lbs, or unit loading of 953 psi. These bearings, however, are capable of unit loads of up to 20,000 psi.

The strength of the rod in transmitting this torque is also of concern. The highest area of stress would be at the rod threads which would have a bending stress on the order of 82,900 psi with a 17,100 in-lbs torque. The tensile strength of the rod steel is at least 125,000 psi.

Realizing that some conservative assumptions were made in estimating the torque, the seal friction should not present any problems. The friction in this interface will be determined experimentally to gain further confidence.

2. Strut Trunnion Seals

The strut design presented in the proposal consisted of a strut rod and strut pivot shaft with a solid non-pivoting joint. Because of concern over misalignment between the roadarm and the strut aggravated by load-induced deflections, a flexible trunnion joint was designed to minimize any external side loading of the strut (see Figure 14). The trunnion joint contains a unique sealing bushing that maintains a tight clearance sealing interface irregardless of position, tolerance stackups, bearing clearances, or bearing wear.

The trunnion provides a rotational degree of freedom perpendicular to both the strut rod axis and the strut pivot shaft axis. It consists of two radial self-lubricated bearings to transmit the load through the joint as well as two brass thrust washers to carry any side load. Each of these bearings must have clearances to allow for tolerance stackups and deflections. These bearings are also subject to wear. The trunnion must communicate and seal two different fluid volumes, the strut piston end and the strut rod end. Each of these passageways communicate to the strut pivot shaft through the ears of the trunnion. The sealing bushing fits inside each of the trunnion ears and provides sealing while allowing freedom of motion in four different directions. A face seal on the bushing allows freedom in a flat plane, or two degrees of freedom. This allows for wear and misalignment of the radial trunnion bearings. A radial seal on the bushing allows linear and rotational degrees of freedom about the trunnion axis. This allows for swiveling of the trunnion joint and axial motion in the joint due to clearances and deflections.

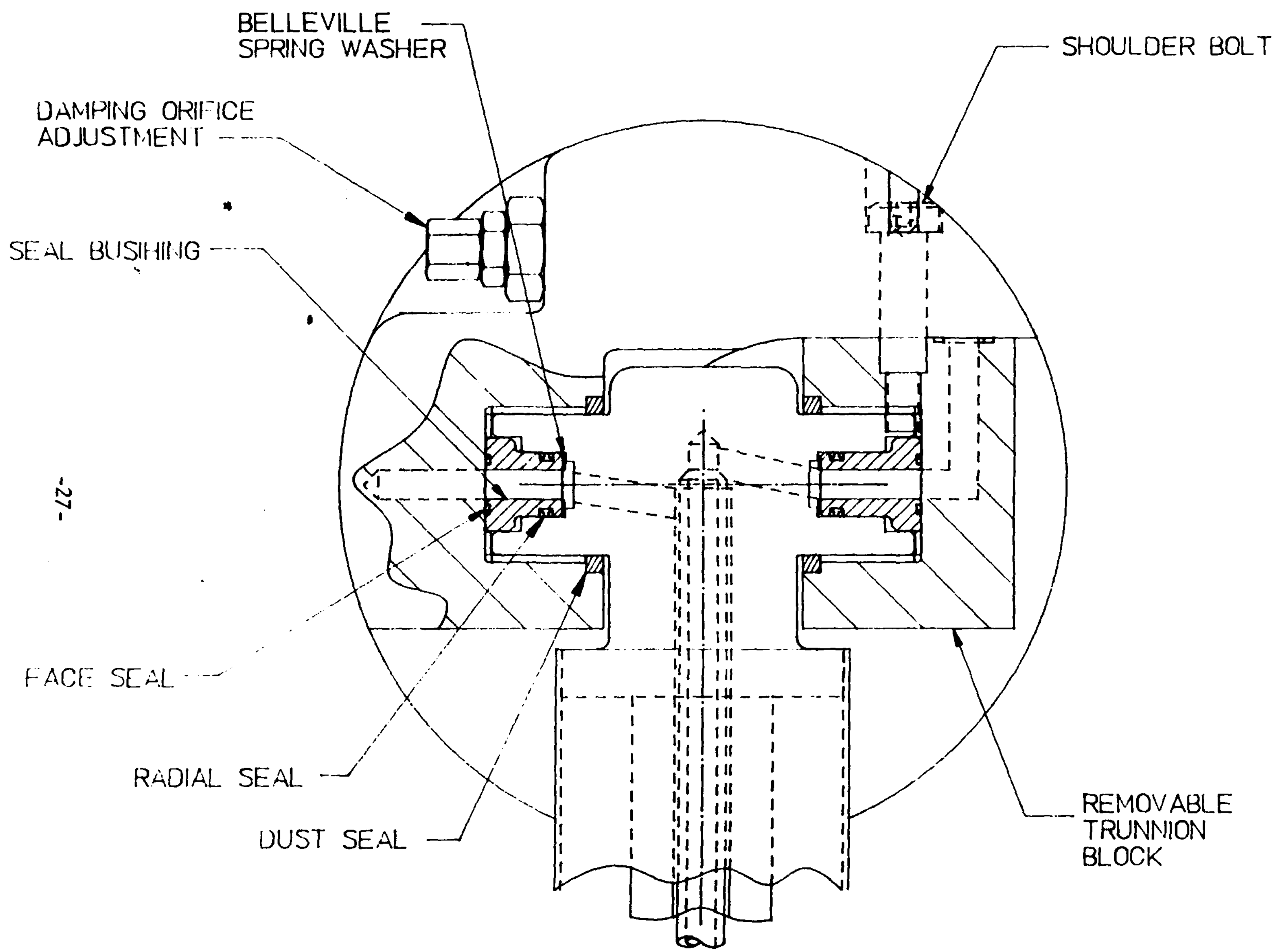


FIGURE 14. TRUNNION JOINT

The two seals on the sealing bushing are sized so that they will be pressure balanced to maintain a tight clearance in the face seal interface. The outside diameter of the radial seal is slightly larger than the outside diameter of the face seal gland. An initial preload is necessary and is provided by belleville washers to close the face seal gap until pressure is built up to load the bushing. It was determined during the testing of the breadboard suspension unit that the bushing must be preloaded with a spring force of at least 50 lbs to prevent the face seal from blowing out. Both of the sealing bushing seals have nylon backup rings to prevent extrusion while subjected to the pressures of up to 18,000 psi.

3. Dust Seals

There are several moving interfaces that must be protected from the elements of dust and sea water. The primary ones are the strut pivot shaft and the roadarm pivot. A mechanical metal face seal was specified for these interfaces such as the CR Industries HDDF seal. These seals consist of two lapped metal sealing rings. An elastomer belleville washer preloads each of the two rings together to provide positive lubricant retention and dirt and water exclusion. The metal rings are made of a high alloy corrosion-resistant steel. The strut trunnion bearings must also be protected from the elements. Since this interface has very little motion, a typical lip seal was specified.

4. Miscellaneous Seals

Other seals contained in this suspension system are all static seals. All of these seals will be o-ring type seals and those that are exposed to high pressures will have nylon backup rings. The low pressure seals will not use backup rings.

E. Damping Analysis

An analysis of the damping characteristics that can be expected in this suspension system design was conducted and a general equation was derived to predict the damping force as a function of the fluid viscosity, the damping orifice area, and the roadwheel vertical displacement.

All of the damping values discussed will be for the suspension system being at the normal static position. Because of the change in the geometry of the strut-roadarm linkage mechanism, the effective damping coefficient can be 38 percent higher at the full jounce position than at the normal static position. The equation derived for the damping force is as follows:

$$F_{RD} = 2.15 \times 10^5 \, V_R + \mu \frac{V_R^2}{3430 \, A_o^2} \tag{2}$$

where:

F_{RD} = vertical damping force at the roadwheel (lbs)

V_R = roadwheel vertical velocity (in/s)

μ = fluid viscosity ($lb\text{-}s/in^2$)
($100 \; cSt = 1.38 \times 10^{-5} \; lb\text{-}s/in^2$)

A_0 = damping orifice area, in^2

The first component of the equation represents the damping force produced just from the pressure drop in the fluid passages. The second component of the equation is the damping force due to the restriction from the damping orifice.

The desired minimum damping coefficient was 70 lbs s/in. With a unit weight of 5,167 lbs and a spring rate of 503 lbs/in, the critical damping coefficient is 164 lb-s/in. Therefore, the minimum design damping ratio is 0.427. At a damping force above 2,800 lbs, the force is limited by a relief valve located in the piston of the strut.

The effect of temperature on the damping characteristics can be described below, where the damping coefficient for the suspension unit is shown assuming no restriction from the damping orifice and assuming that the Dow Corning 200 fluid, 350 cSt viscosity grade is used.

Table 4. Fluid Temperature Effects on Damping With No Orifice Restriction (Rebound Damping)

Temperature (°F)	Viscosity (cSt)	Damping Coefficient (lb-s/in)	Damping Ratio
-50	2000	59	0.36
0	900	26.7	0.163
100	260	7.71	0.047
250	85	2.52	0.015

The above damping ratios would apply to the suspension unit during rebound as the flow bypasses the orifice.

Note that in Table 4 the viscosity of the silicone fluid is relatively insensitive to temperature. A hydraulic fluid with a similar viscosity at 0°F of 1000 cSt (150 SSU Grade) has a viscosity of 32 cSt at 100°F and 40,000 cst at -50°F.

The sensitivity of the damping adjustment is shown below for a roadwheel vertical velocity of 20 in/s.

Table 5. Damping Adjustment Characteristics.

Adjustment Turns Out from Setpoint	Orifice Area	Orifice Damping Ratio	Combined Damping Ratio at 100°F
1	.0063	.89	.94
1.5	.0093	.410	.46
2	.0120	.25	.30
3	.0167	.13	.18

Therefore, the desired nominal orifice setting will be approximately 1.5 turns out. The damping orifice and check valve design is illustrated in Figure 15. To set the orifice, however, a reference setpoint must first be determined. To do this, the adjustment must be turned completely in until it bottoms out. The setpoint will be exactly 4 turns out from the bottomed out position. The correct adjustment can then be made from this setpoint.

The damping characteristics are plotted in Figure 1C. A curve is plotted for three different damping adjustments at a temperature of 100°F which fall in the range specified by DTRC. The nominal setting where the screw is set, 1½ turns out from the setpoint, is also plotted for a temperature of 0°F. The curves are nonlinear due to the nonlinear relationship of the damping orifice. For each setting, the curve rises with velocity until a force level of about 3000 lbs is reached. At this level, the damping relief orifices limit the damping force.

The damping relief orifice is located in the strut piston (Figure 16). It consists of a differential area poppet valve that allows flow through its center to the piston end of the strut. It allows flow to relieve from the piston end to the rod end when a differential pressure of 4000 psi is reached. To accomplish this, the seating diameter is slightly larger than the shank diameter of the poppet. When 4000 psi of pressure is applied to this annular area, it will overcome the preloaded force of the spring, lifting the poppet off the seat.

F. Strut Design

The strut design uses the basic technology developed by Liquid Spring Corporation for use in suspension struts provided for heavy haul trucks for the mining industry. Typically, the struts manufactured are self-contained; all of the necessary fluid volume is contained in the cylinder, as well as all of the damping orifices. A layout of the Liquid Spring strut for this suspension system is shown in Figure 17.

The design had to be modified for this application to provide connections from both the rod and piston ends of the cylinder by means of passageways internal to the rod and through a trunnion-mounted rod end to the strut pivot shaft. The rod gland area was redesigned and streamlined to minimize the diameter and length. A cushion absorbs the shock loads that may be encountered when the strut reaches full extension. A seal on the piston provides a positive seal between the rod end and the piston end.

Some of the more salient features of the strut include special high-pressure rod seals, which are unique to the Liquid Spring design. There are bearings for the rod and piston to carry any lateral loads applied to the assembly. A boot connected to the rod end and covering the rod and part of the cylinder provides protection for the rod and also enhances the dissipation of heat by pumping air across the cylinder as the rod reciprocates.

The strut also incorporates a relief valve (not shown in Figure 17) in the piston end of the rod as described in the previous section.

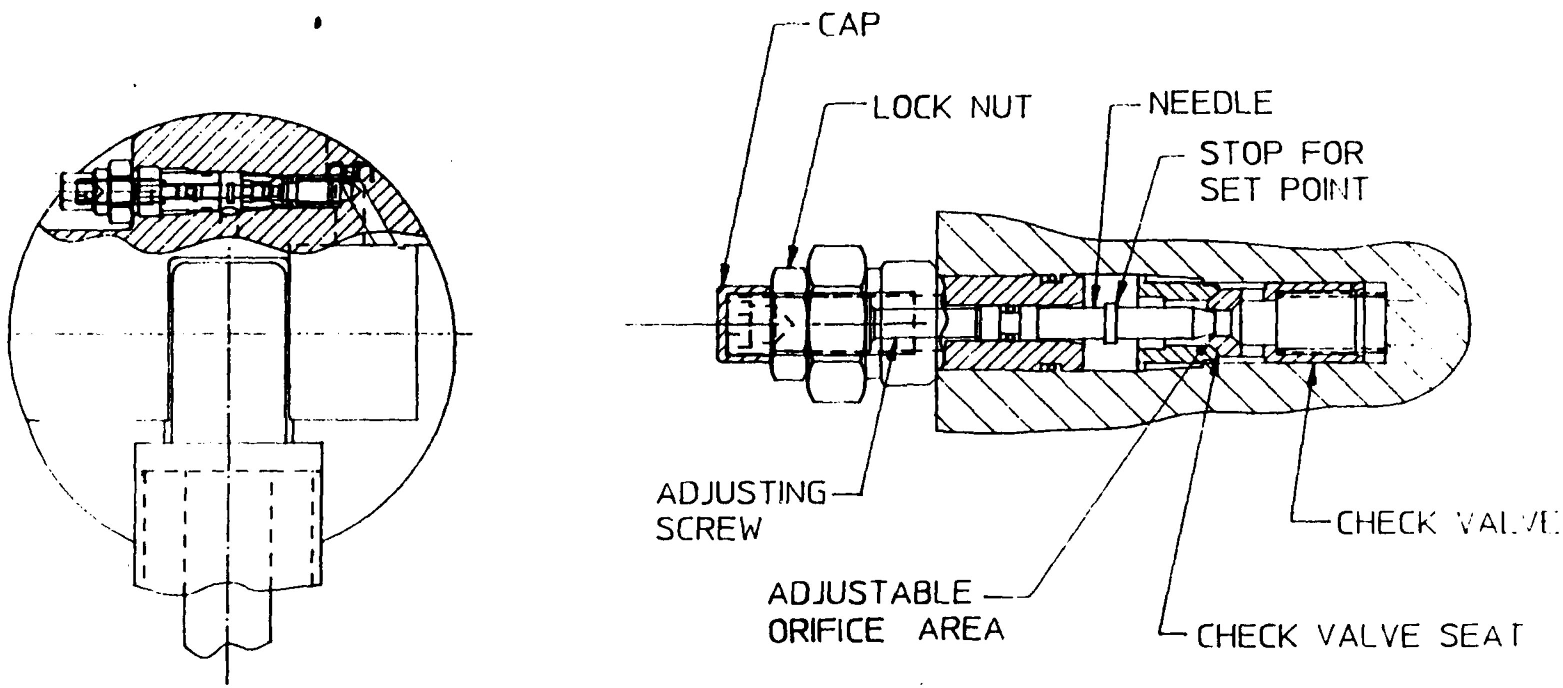


FIGURE 15. DAMPING ORIFICE AND CHECK VALVE

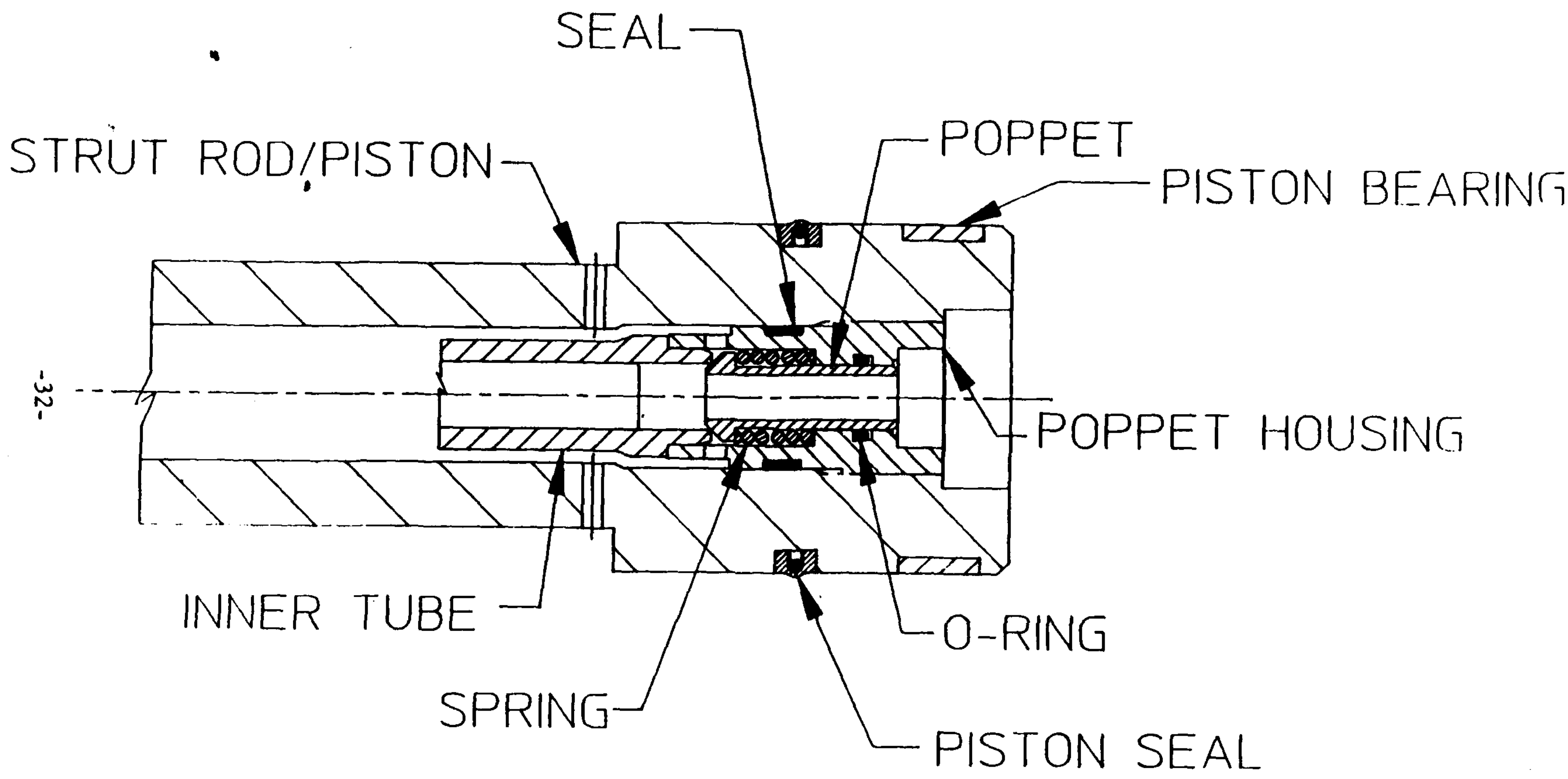
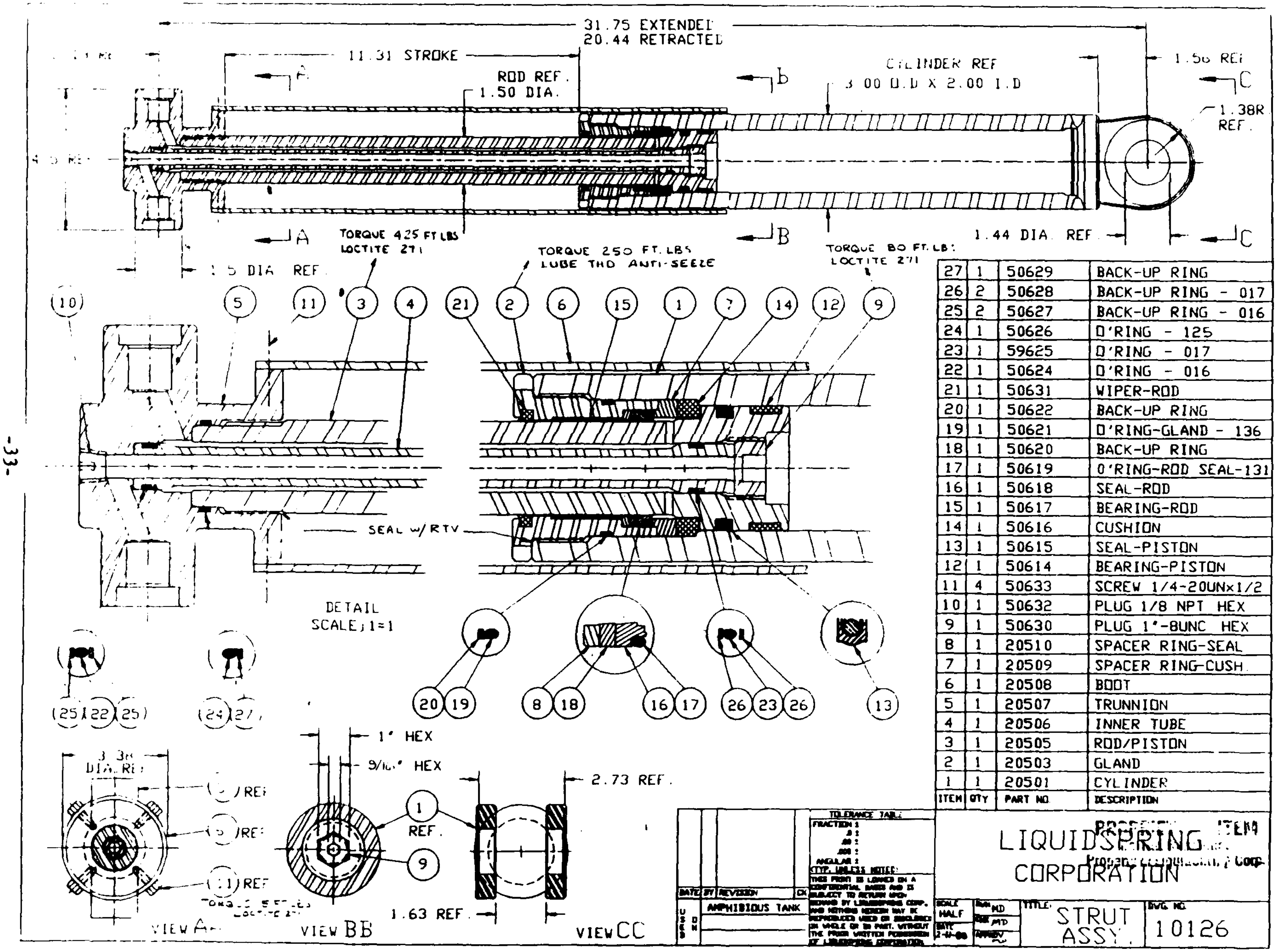


FIGURE 16. DAMPING RELIEF VALVE



ITEM	QTY	PART NO.	DESCRIPTION
27	1	50629	BACK-UP RING
26	2	50628	BACK-UP RING - 017
25	2	50627	BACK-UP RING - 016
24	1	50626	O'RING - 125
23	1	59625	O'RING - 017
22	1	50624	O'RING - 016
21	1	50631	WIPER-ROD
20	1	50622	BACK-UP RING
19	1	50621	O'RING-GLAND - 136
18	1	50620	BACK-UP RING
17	1	50619	O'RING-ROD SEAL-131
16	1	50618	SEAL-ROD
15	1	50617	BEARING-ROD
14	1	50616	CUSHION
13	1	50615	SEAL-PISTON
12	1	50614	BEARING-PISTON
11	4	50633	SCREW 1/4-20UNx1/2
10	1	50632	PLUG 1/8 NPT HEX
9	1	50630	PLUG 1"-BUNC HEX
8	1	20510	SPACER RING-SEAL
7	1	20509	SPACER RING-CUSH.
6	1	20508	BOOT
5	1	20507	TRUNNION
4	1	20506	INNER TUBE
3	1	20505	ROD/PISTON
2	1	20503	GLAND
1	1	20501	CYLINDER

FIGURE 17. LIQUID SPRING SUSPENSION STRUT
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G. Control Valve

The control valve for controlling the extension and retraction of the suspension units is purely hydromechanical, and requires no electronics. During normal operation the control valve does not affect the suspension unit because the suspension units are completely self-contained and passive, requiring no other energy input. The control valve transfers fluid to or from the suspension units only during extension or retraction or after the suspension units have changed their operating temperature considerably.

The basic purpose of the control valve during the extend mode is to allow the proper volume of fluid to be charged into the system. The volume can be determined by measuring the pressure if a unit is at a known position. However, all units will not be at the same position. If the system is extended in water, the front and rear units will be more compressed than the middle units due to the track tension. If the system is extended on land, the surface will probably be uneven so that all of the units will be at different positions. Thus, it is necessary for the control valve to sense both pressure and position for determining its position.

Figure 18 is a cutaway drawing of the control valve parts as they exist with the suspension system in a normal static condition. The control valve system consists of several valves. The spool valve is a four-way, three-position directional control valve. A pilot-operated check valve isolates the spool valve from the suspension system, preventing the suspension system from leaking down and prevent the high pressures encountered at full jounce positions from acting on the spool valve. In the strut pivot shaft there is the damping orifice and check valve as well as a pilot-operated check valve for disabling the damping orifice and check valve during the retract mode. The relief valve in the strut piston is involved in the control circuit only from the standpoint that it must provide a positive seal between the two volumes in the strut to prevent the suspension unit from creeping down while in the retracted mode.

The spool valve senses the strut angular position and the pressure in the suspension unit to determine its position. As long as the proper pressure exists for any angular position of the strut, the spool valve will remain in a neutral position which does not affect the suspension unit. A cam located on the strut pivot shaft, indexed to the bearing locking nut keyway, displaces a cam follower that compresses a spring. The spring forces the spool valve down so as to permit flow of fluid from the pressure supply to the suspension system (Figure 19). The pressure supplied by the suspension unit from the control valve is exposed to an annular area on the spool valve. The resulting force opposes the cam spring to move the spool valve back to the neutral position as the proper pressure is built up for that particular suspension unit's position. The profile of the cam is such that it will create a spring force in proportion to the proper pressure for any strut position.

If the suspension system fluid temperature rises considerably, due to change in climate or the generation of heat from damping, the fluid will expand and cause the unit to extend. Without the control valve, the suspension system would change its position one inch for every 10°F change in fluid temperature. If the position of the unit does rise, the cam force will be less than the pressure force

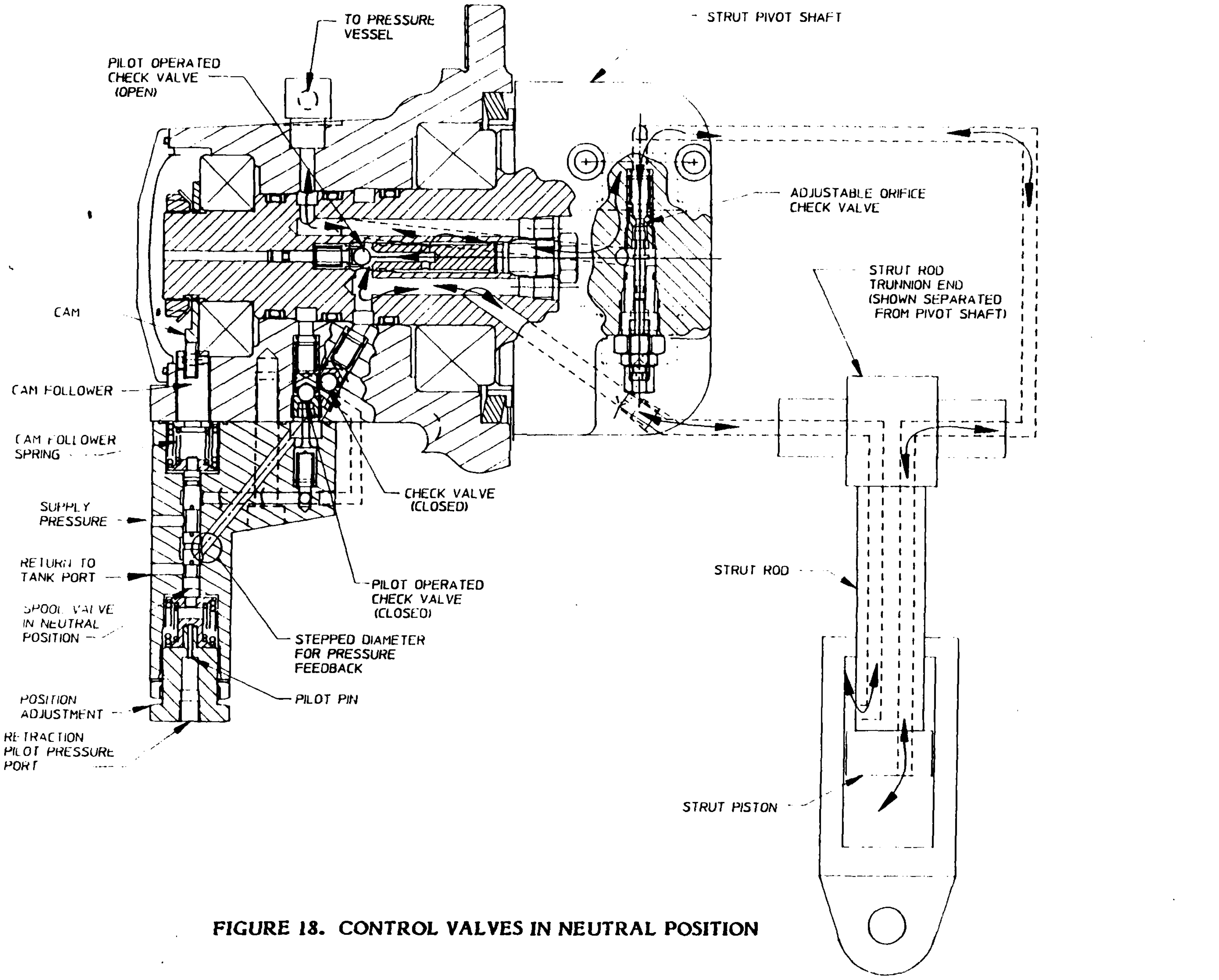

FIGURE 18. CONTROL VALVES IN NEUTRAL POSITION

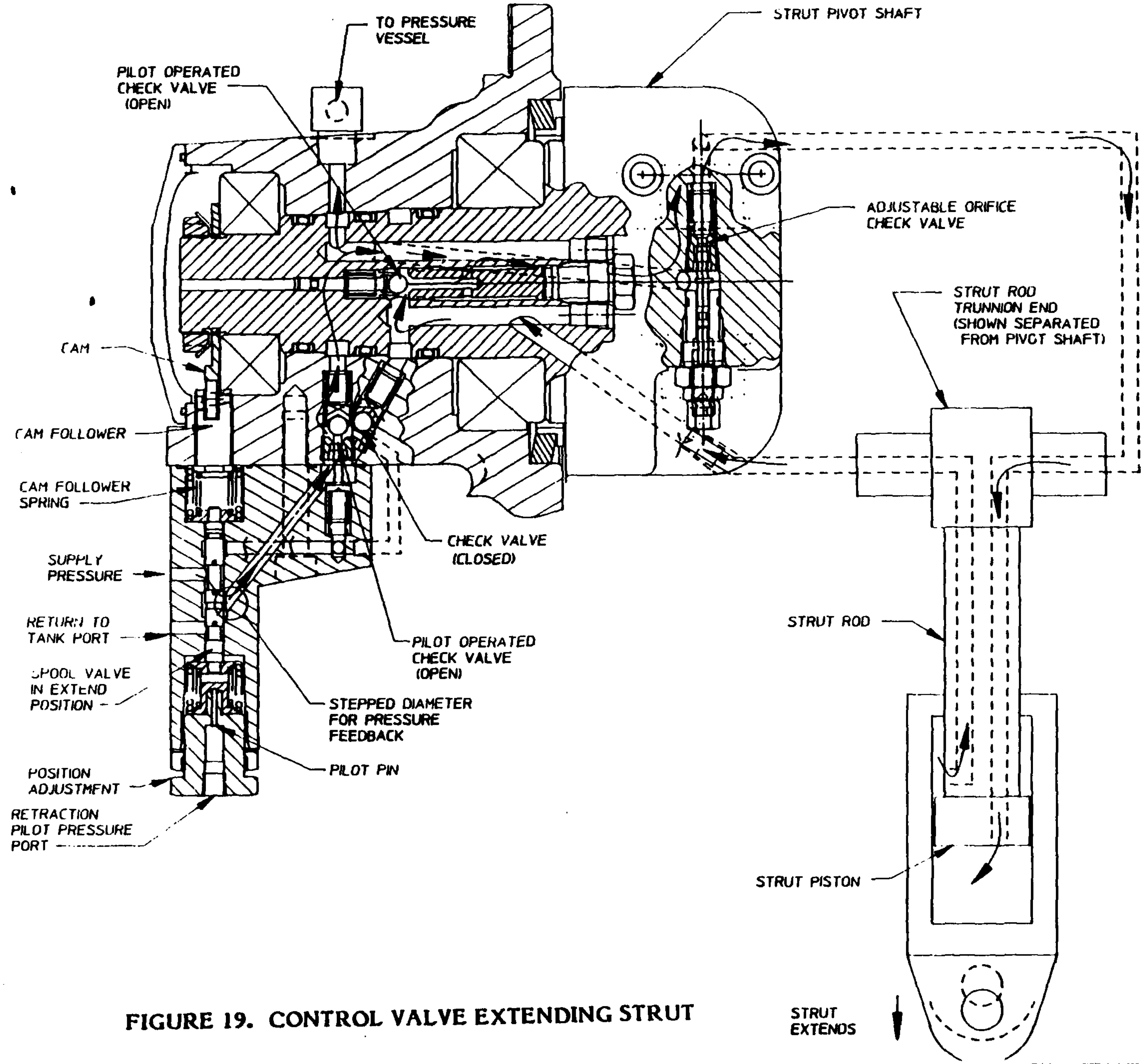

FIGURE 19. CONTROL VALVE EXTENDING STRUT

allowing the spool valve to move upwards, porting supply pressure to the rod end of the struts as if to retract the unit (Figure 20). At the same time, the pressure supplied by the spool valve activates a pilot-operated check valve, allowing fluid to bleed from the piston end of the strut, through the spool valve and back to the main system fluid reservoir. When the unit's position matches the correct pressure, the valve will again return to neutral.

To retract the suspension system, a pilot pressure supply must be activated to the suspension units. This pilot pressure acts on a small piston that compresses a spring on the bottom end of the spool valve. This force adds to the force exerted by the suspension system pressure on the differential area of the spool. The spool shuttles upward as to bleed some of the fluid off as just described. The force from the pilot pressure will hold the spool valve in this upward position. If the vehicle is in the water, the fluid will bleed off until the pressure in the unit is below 500 psi. At this time, a pilot-operated check valve will close in the strut pivot shaft blocking flow through the damping orifice and check valve. Flow from the spool valve will continue to the rod end of the strut, retracting the strut as fluid from the piston end of the strut continues to bleed off through the spool valve.

When retracting the suspension units on land, the sequence of events is the same except that the vehicle will lower itself to the ground due to its own weight as fluid is bled off. When the load is taken off the suspension units, the pressure then starts to drop. When the pressure drops below 500 psi, the pilot-operated check valve closes and the rod end of the strut is pressurized, which completes retraction of the unit.

As long as supply pressure is provided to the suspension unit, pilot pressure must also be supplied to maintain the unit in the retracted position. If there is no supply pressure to the units, they will remain in the retracted position. Without supply pressure the units cannot be extended or retracted. For instance, if one unit is damaged and loses its fluid, or if the fluid supply system is disabled, or if the engine stalls, the remaining suspension units will maintain their setting and continue to function. A hand pump will be provided to charge the units in case of failure of the pressure supply system.

H. Pressure Vessel

The pressure vessel for the suspension unit is required to house the additional compressible volume not contained by the strut. To minimize the weight of the pressure vessel it was decided to investigate the use of composite materials. To manufacture a contained out of steel to withstand pressures of 18,000 psi repeatedly, and with a volume of 265 cubic inches, the weight of the container alone would be in excess of 100 lbs. Using composite materials it is possible to bring the weight down below 25 lbs.

Two companies were contacted in regards to providing this pressure vessel. The one selected was ABB Composites of Irvine, CA. The proposal including a sketch of the pressure vessel is included in Appendix C.

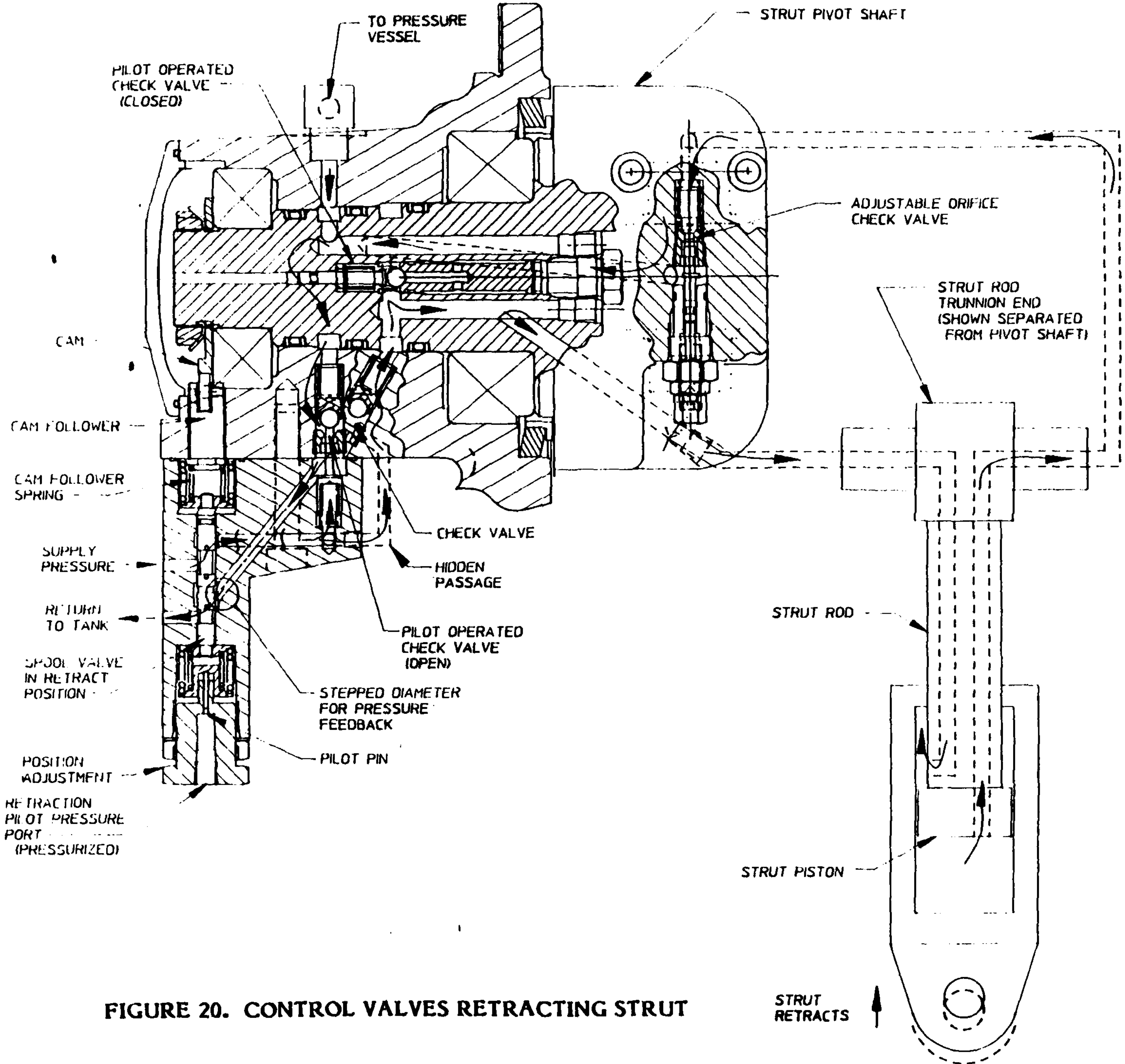

FIGURE 20. CONTROL VALVES RETRACTING STRUT

The pressure vessel (or bottle) would be constructed with a spun aluminum liner that has an approximate wall thickness of 0.18 inch. The liner would be wound with high strain carbon fiber to an approximate thickness of 0.5 inch. The overall outside diameter would be approximately 7.5 inches and the overall length would be about 15 inches. The bottle would have a rated operating pressure of 17,000 psi which means that it could cycle up to that pressure on the order of 500,000 times without failure. The rated minimum burst pressure would be 28,000 psi. These pressure ratings are for temperatures of 75°F. At 250°F the strength would be reduced about 28 percent. These ratings should adequately handle the requirements of the system.

I. Corrosion Protection

Since this suspension system will be in and out of sea water, corrosion of all the exposed parts is of major concern. In particular, the integrity of the strut rod must be maintained. Typical chrome plating as normally used on strut and hydraulic cylinder rods will pit and peel off in this type of environment. Marine life will also adhere to chrome. The positive features about chrome are that it provides a hard, smooth, and low friction surface upon which seals operate very well. Since chrome plating is an electrochemical process it also has a very good bond strength to the parent material.

In a search for an alternate coating to chrome, spray on and plasma coatings were ruled out because they do not possess adequate bond strength. They are bonded to the parent material only mechanically. Plasma spray coatings offer the best bond strength at about 10,000 psi. This would be insufficient since the strut rod would be exposed to pressures up to 18,000 psi. A crack in the coating would allow fluid to migrate between the coating and the parent material, potentially separating the coating from the metal.

A type of spray-on coating that would have a higher bond strength is a spray and fuse coating. In this process the coating is sprayed on and then the whole part is heated up to fuse the coating to the parent material. This fusing operation would destroy the strength properties of the steel as well as warp the part.

The process selected for coating the strut rod is called "shearadizing" by Lovatt Technologies. With this process the part is finished to the correct dimensions and surface finish before processing and requires no finish grinding. This is an electrochemical process performed at subfreezing temperatures that does not add any material to the part but rearranges the molecular structure of the surface of the part. The result is a very corrosion-resistant surface, especially if the parent material is basically corrosion resistant. The surface hardness is also increased on hardenable steels from 10 to 20 points on the RC hardness scale. The surface is also lower in friction and more abrasion resistant. The cost of the process is comparable to that of chrome plating. The appearance is not the shiny finish of chrome, but a dull finish similar to the color of the parent material. The material selected for the rod is 17-4PH stainless steel, which has very good strength properties as well as corrosion resistance.

A similar process by the same company for aluminum is called "banadizing." This process will be used on the suspension system housing which is made of a high strength aluminum alloy. Banadizing will not only increase the corrosion resistance of the housing but serve as an excellent mating surface for the roadarm pivot bearings and the strut pivot seals. Hard anodizing is highly recommended as a mating surface for the DU self-lubricating bearings of the strut pivot. Banadizing is much better than hard anodizing for several reasons. Hard anodizing can change the ductile and fatigue properties of the material by up to 60 percent. With banadizing these properties are not affected. Hard anodizing is very brittle, banadizing is not. Banadizing is also extremely hard, on the order of 90 on the RC scale.

Unlike shearadizing, banadizing adds material to the surface. For a 0.002-inch coating, half will be in the parent material and half will be added material.

J. Fabrication and Test Plan

Breadboard testing is underway and fabrication of the first two of the required 14 delivered suspension units will begin shortly. As production parts are made available they will replace parts on the breadboard test.

The test plan for the breadboard design is to accumulate hours of cyclic testing. Teardown and inspection of the units will take place when it becomes appropriate depending on the progress of the test. When the first two production units become available they will replace the breadboard unit and endurance testing will continue for at least 200 hours on each.

The endurance test will consist of a test cycle as listed below:

- Continuous automatic cycle (20 cycles/minute), 3 inches below to 3 inches above normal static
- Hourly:
    - 10 cycles, 3 inches below to 8 inches above normal static
    - 3 cycles, 3 inches below to 16.5 inches above static
    - 5 cycles, 10,000-lb side load from outside
    - 5 cycles, 10,000-lb side load from inside
    - Log temperatures and pressures at static position

For each of the following twelve production units, this cycle will be run for 10 hours.

K. Vehicle Interface Requirements

The suspension units must interfere with the vehicle with its mounting pad and with its fluid power supply to the control valves for extension and retraction.

The mounting hole and bolt pattern are shown in Figure 21. Access holes from inside the vehicle are required for installation, removal, and service of the suspension units. These hole locations are shown relative to the hull opening in Figure 22. From these access holes the hose connections can be reached, the control valve can be adjusted, and the system can be bled of any trapped air. The upper hole in the plan view will allow access to the inner bearing of the strut pivot.

The supply of fluid to each suspension unit must tap into three different lines: a high pressure line supplying fluid to extend the units, a medium pressure pilot line for signalling the units to retract, and a low pressure line for returning fluid back to a central reservoir from the control valve during the retraction operation, from the air bleed, and from any valve or seal leakage. The high pressure line must be rated for at least 7500 psi operation and be capable of carrying a flow of up to 1 GPM. The low pressure line should be rated for at least 100 psi and have the same flow capacity. The pilot pressure line should be rated for 4000 psi and can be of the smallest size available since it is just a sensing line.

The specifications of the supply pump depend on the desired behavior and response of the extension and retraction controls. As a minimum, the pump must be rated for a pressure of at least 6000 psi. This will charge all of the units to the correct static position if the vehicle is on flat level ground. With a pressure of 7500 psi available, a roadwheel may be as much as 2.5 inches out of position with the rest of the roadwheels and still be correctly charged. A pressure supply of 7500 psi will also allow the vehicle to be raised with a variation of the load distribution of up to 25 percent from front to rear or side to side. The units with the least amount of load will extend first because the fluid will flow to the units that provide the least amount of resistance.

The flow capacity of the pump will depend upon the desired speed for extension and retraction. The amount of fluid required to extend all twelve units is 400 cu. in. To retract them requires a pump flow of 165 cu. in. The reason the extension flow is higher is because the units must not only be displaced but they must be precharged by the pump. To retract, the control valve bleeds off the precharged volume to the tank and the pump only has to displace the rod volume to retract. Thus, if it is desired to extend the units in 15 seconds a flow rate of 6.9 GPM is required. At this flowrate, retraction will only take 6.2 seconds.

It is desirable to have a pump that is pressure compensated so that a continuous pressure is available. This will insure the proper setting regardless of the fluid temperature. Dynex Rivett manufactures a variable displacement axial piston check valve pump that has a displacement of 1.8 cu. in./rev. At a shaft speed of 1800 rpm the output would be 12.3 GPM. It is rated for a maximum pressure of 8000 psi. Whether it is compatible with silicon fluid must be confirmed with the manufacturer. Other pump options include fixed displacement pumps that are only activated during extension or retraction or when a correction in the ride height is desired. Another option would be to use a booster or intensifier powered by an onboard air source or hydraulic source.

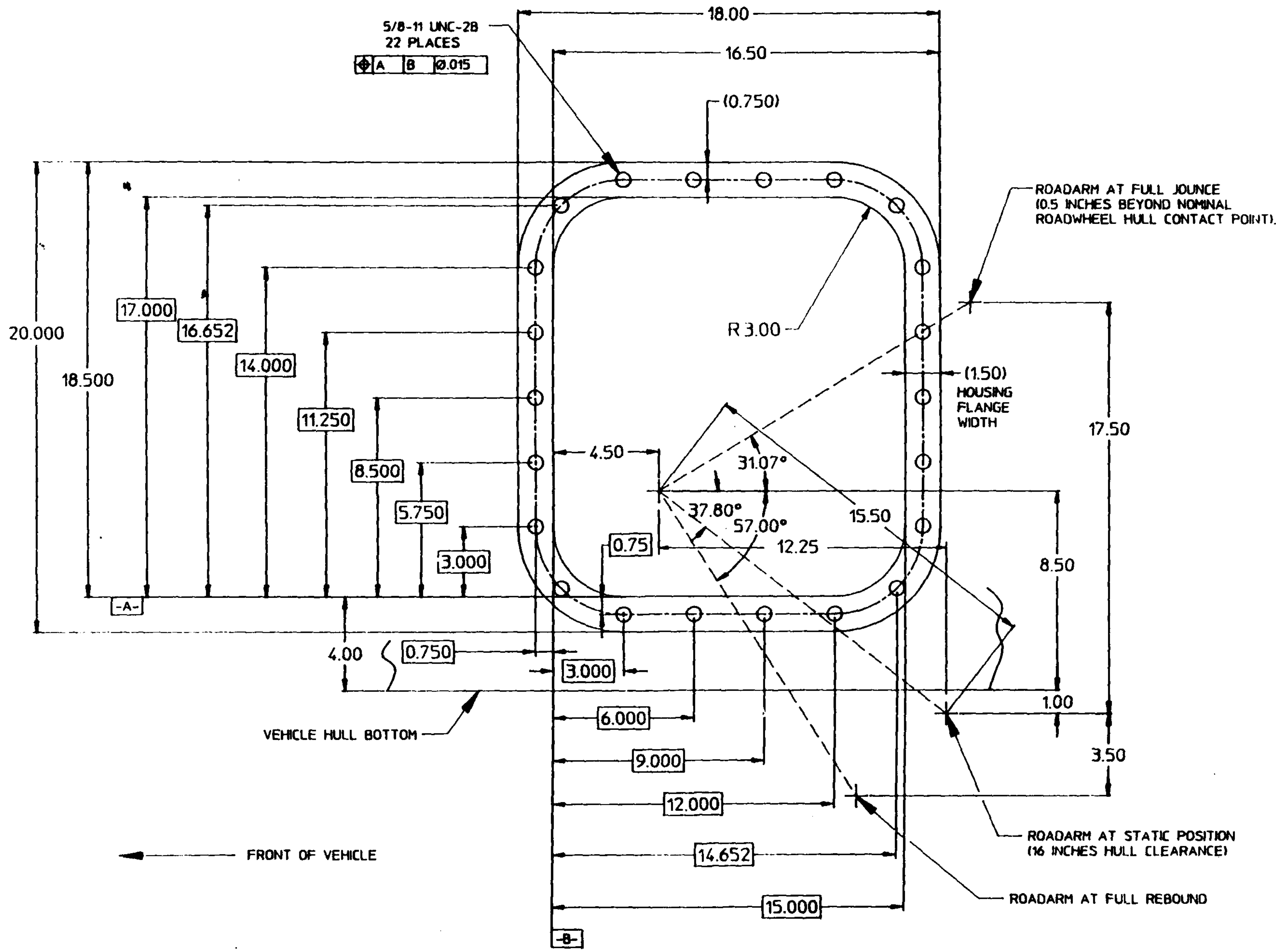

FIGURE 21. VEHICLE MOUNTING INTERFACE

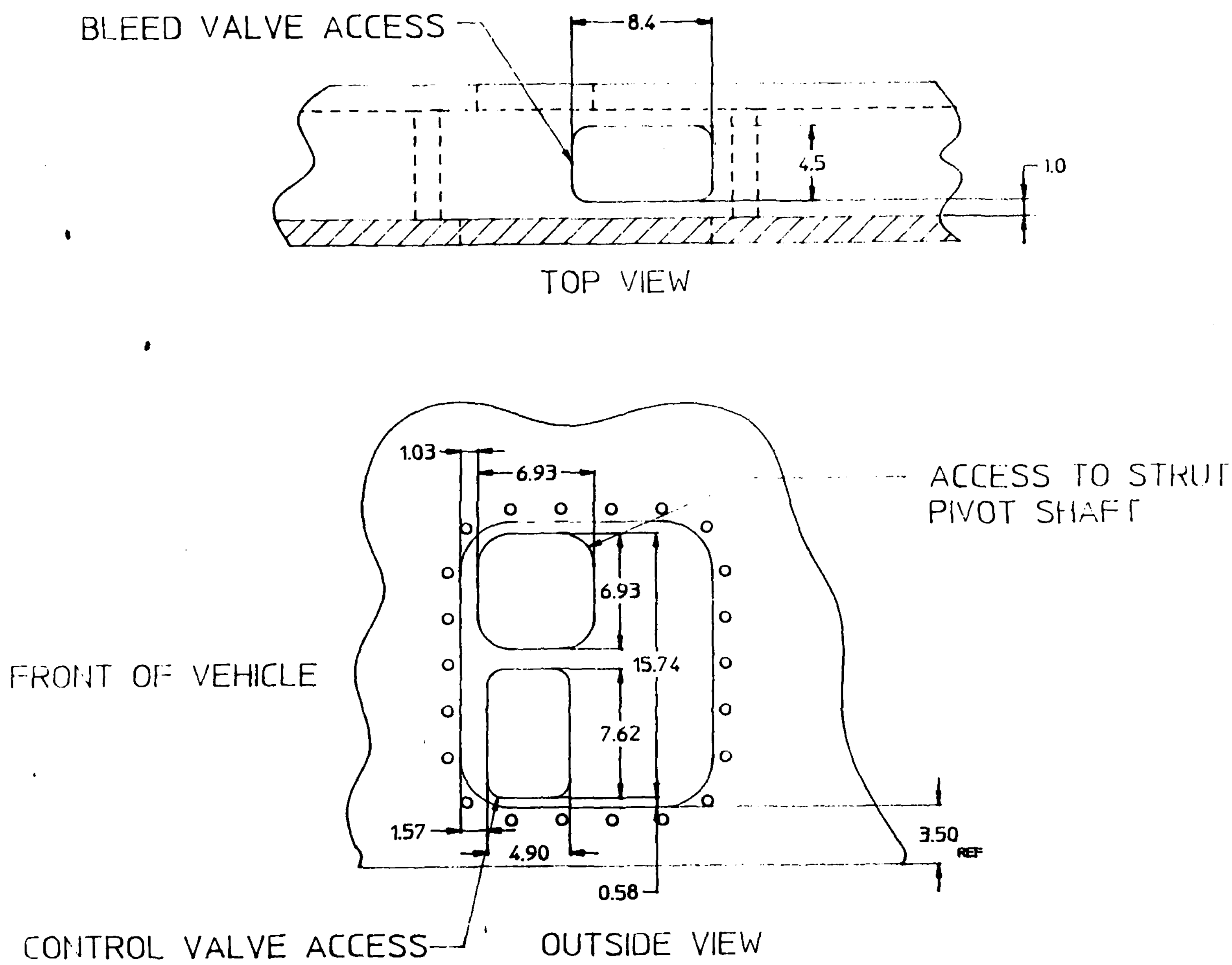

FIGURE 22. ACCESS HOLE REQUIREMENTS

The capacity of the storage reservoir should probably be about 3 times the maximum amount of fluid pumped, or 1200 cubic inches, or about 5 gallons. An on-off switching valve and a pressure reducing valve will also be required for the retraction function as illustrated in the system schematic shown in Figure 23. A filter on the return line to the reservoir is also recommended.

L. Projected Weight

The projected weight for each suspension unit is approximately 242.3 lbs. The total weight for 12 units would be 2,908 lbs. The materials for all of these parts are steel except for the housing and roadarm, which are aluminum, and the pressure vessel which is a composite material. The weights of the major individual parts and assemblies are itemized below. Additions to these weights are listed if the roadarm and housing are made of steel also.

Table 6. Projected Weights

Each Suspension Unit	Final Design (lbs)	Steel Option (lbs)	Proposed Estimate (lbs)
Roadarm	24.0	+30	51.12
Strut	58.0		63.22
Strut Pivot Shaft	54.0		
Housing	50.1	+79.9	135.67
Pressure Vessel	25.0		
Control Valve	11.3		15.7
Spindle and Hub Assembly	19.0		19.0
Roadarm Strut Pin	1.3		3.96
Mounting Bolts	4.5		6.76
Total per Unit	247.2	+109.9	295.43
Total per 12 Units	2966.4	+1,319	3545.16

Fluid Supply System Estimates	Final Design (lbs)	Steel Option (lbs)	Proposed Estimate (lbs)
Fluid in suspension units (16.3 gal)			
Fluid in lines (0.45 gal)*			
Fluid in reservoir (3.5 gal)			
Total Fluid (20.25 gal)	161		309
Reservoir (5 gal)	16		110
Filter	3		3
Pump	140		140
Lines, Hoses, and Valves	66		65
Total Suspension System Weight	3,351	+1,319	4,172

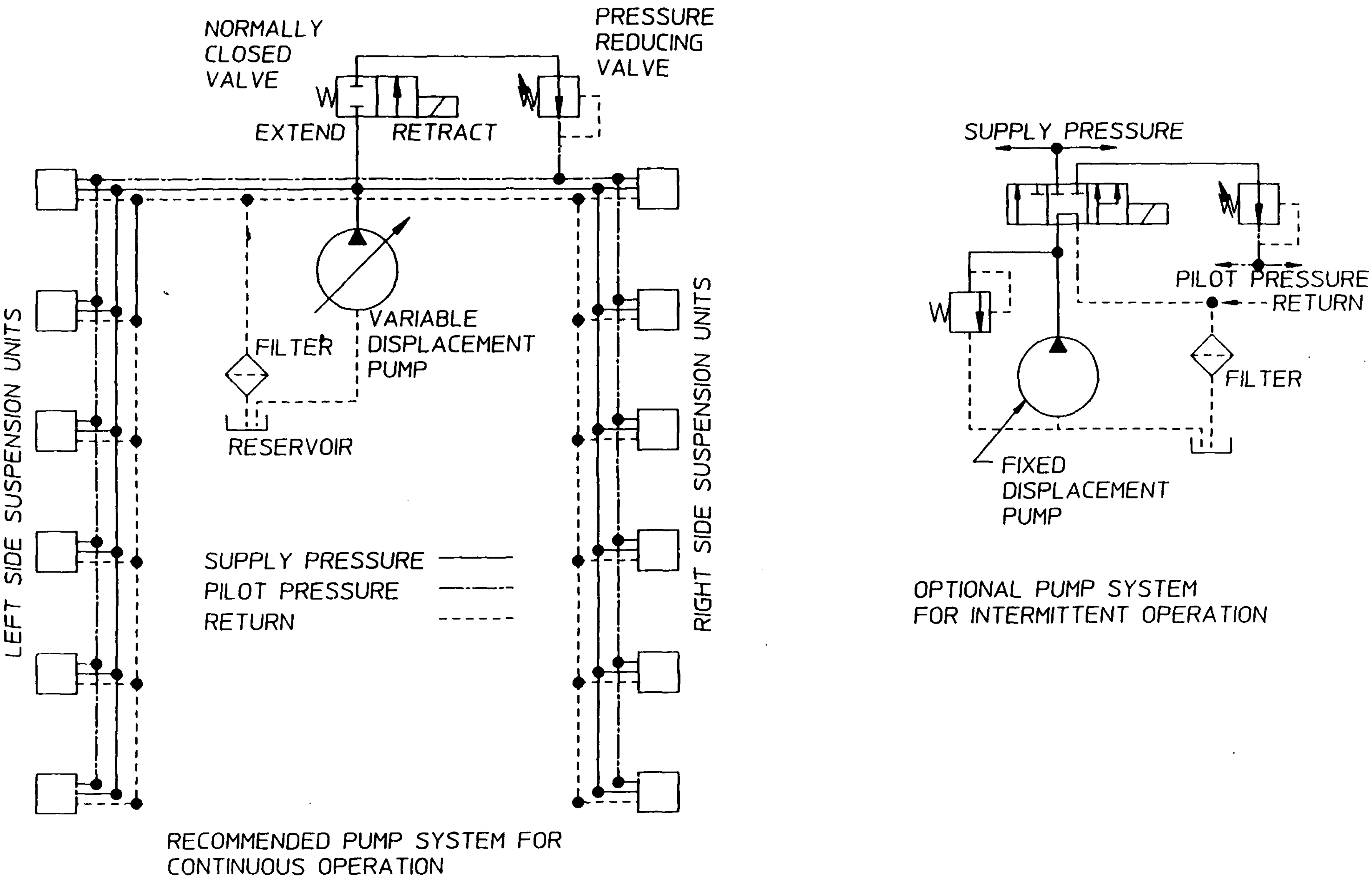

FIGURE 23. VEHICLE FLUID INTERFACE

The weights using an aluminum housing and roadarm are well below the weight projected in the proposal of 295.43 lbs each. Using aluminum on the roadarm and housing may present some marginal strength situations, especially if the worst case load conditions as modeled are very likely to be experienced. The strength of the roadarm pivot shaft can be improved by incorporating a tapered steel shaft in the aluminum housing. The unit weight would increase by 15.3 lbs to 262.5 lbs, still below the proposed weight. If the strength of the roadarm is in question, it could instead be made out of steel along with the steel shaft for a total unit weight of 292.5 lbs, right at the proposed unit weight.

The course of action at this time is to manufacture the housing and roadarm out of aluminum.

M. Maintenance

Periodic maintenance on these units is estimated to be rather minor. The area that will require attention is the spherical bearing that connects the strut to the roadarm. The bearing has seals on the spherical ball; however, when operating in sea water, water will inevitably get into the ball area and all the interfaces with the pin. It will thus be necessary to keep this interface well greased, particularly after operation in sea water.

A filament-wound bearing, which does not require any maintenance, will be retrofitted and tested on the breadboard unit, and will replace the spherical bearing if demonstrated to be acceptable.

The seal for the roadarm pivot will also require some periodic maintenance. It is a metal-to-metal face seal and should have a film of light oil (SAE 30) or less to keep it lubricated. This will lubricate the DU self-lubricating bearings, adding to their life. Heavy oil or grease should not be used since cavitation damage may result in the DU bearings as a result of a high viscosity lubricant and any high rotary velocities. A plugged oil fill hole is provided for checking the level and adding lubricant.

The DU bearings in the strut trunnion will also require greasing occasionally, mainly to keep the salt water purged from this interface. Grease fittings will be provided on the strut pivot shaft.

If for any reason the system gets air in it, the air must be purged from the suspension units to maintain the proper spring characteristics. If there is an external leak and the fluid level is low in the reservoir, the pump will start sucking air and forcing it into the system. The silicone fluid must be maintained at the proper level. Air can be bled from each suspension unit by an air bleed valve provided at the top of the pressure vessel. The air and fluid will be ported back to the reservoir, avoiding any spills inside the vehicle.

IV. BREADBOARD BUILD AND TEST

A feasibility demonstration suspension unit (breadboard unit) has been constructed and tested successfully. The purpose of building this unit was to demonstrate the concept and also to test some of the critical interfaces of the design. The primary areas of concern are the seals in the strut pivot shaft and the trunnion joint. Other areas of concern were the strut, the roadarm pivot, and the strut pivot bearings. The breadboard unit was much like the production design except that no attempt was made at minimizing the weight and there were not control valves. All of the structural parts were machined out of blocks of steel or welded steel plates. The pressure vessel was made of a heavy steel tube with end caps.

The units was operated for several hours when a seal in the strut pivot shaft blew out. This failure was secondary to a failure of the inner tapered roller bearing on the strut pivot shaft. The tapered roller bearing failed due to an interference problem between its cage and the housing. The failed bearing subsequently caused a failure of the shaft locknut washer which allowed the locknut to back off several turns. The resulting excessive endplay allowed the outermost strut pivot seal to get out of place axially and to extrude. The excessive endplay also allowed the strut pivot shaft to become misaligned and wear a groove into the bore of the housing. Another secondary failure occurred in the strut. Due to the tremendous amount of torque required to turn the shaft while the bearing was failing, the strut rod and piston bearings had to carry excessive loads, resulting in galling and wear in the strut bore and on the rod.

The parts were reclaimed and aré reassembled to resume testing. Extra clearances were provided for the bearing cage. The bore of the housing was also ground to a finer surface finish in the area of the seals to improve their wear resistance and to reduce friction.

The thrust load capacity of the bearing that failed was marginal at the maximum load condition. A bearing with twice as much thrust load capacity was located and will be used in the production design.

Friction in the strut pivot interface is a concern. Although calculations of friction predict that the strut should be able to withstand the bending load on the rod, additional tests will be run to determine the friction at different pressure levels. One noticeable result of excessive friction is that the rod deflects as the unit strokes up and down. The deflection of the rod caused the strut boot to misalign with the cylinder and to drag on the cylinder. Since the parts have been reclaimed and the system reassembled, the rod still deflects, but not enough to make the boot drag on the cylinder.

Over seven more hours of cyclic operation had been accumulated when another failure occurred similar to the first. The inner strut pivot shaft bearing again failed, which allowed the outer seal to fail. There was no damage to the shaft or bore, due to misalignment, and the strut was also in good condition. The cause of this failure is due to the poor lubrication properties of the silicone fluid on highly loaded, metal-to-metal interfaces. The bearing in this assembly was the same type as the first assembly.

To correct the problem, the bearing cavities will be packed with an extreme pressure grease and the cavities will be vented to the vehicle interior. Timken Bearing Co. recommended a grease or an EP grease or oil with a Timken EP, ASTM D2782, with a number of at least 35. For the next build, an EP grease with a Timken EP number of 70 will be used. If an oil is preferred, one with a Timken EP number of 75 is available.

The original intent of using the silicone fluid as the bearing lubricant was to minimize the number of lubricants and to provide containment for any leaks out of the strut pivot shaft seals. These cavities were to be vented to the silicone fluid reservoir through the control valve return to tank line.

With the bearing cavities packed with grease, they will be vented to the atmosphere on the interior of the vehicle. The risk is that if any leakage occurs, it will overflow the cavities and accumulate on the floor of the vehicle. Another potential problem is that small amounts of seepage of silicone fluid into the bearing cavities will mix with the grease, and reduce its effectiveness. However, in the seven hours of running, to date, no silicone fluid leakage was observed.

The accumulation of silicone fluid in these bearing cavities will be monitored in the test program to follow. This will provide further information as to recommended maintenance of repacking these bearings.

APPENDIX A

Dow Corning 200 Silicone Fluid

Information about Silicone Fluids

DOW CORNING

DOW CORNING® 200 Fluid, 50 cSt.
DOW CORNING® 200 Fluid, 100 cSt.
DOW CORNING® 200 Fluid, 200 cSt.
DOW CORNING® 200 Fluid, 350 cSt.
DOW CORNING® 200 Fluid, 500 cSt.
DOW CORNING® 200 Fluid, 1000 cSt.

DESCRIPTION

DOW CORNING® 200 fluids, 50-1000 centistokes (cSt.), are medium viscosity polydimethylsiloxane polymers manufactured to yield essentially linear polymers with average kinematic viscosities ranging from 50 to 1000 cSt.

COMPOSITION

Linear polydimethylsiloxane polymers characteristically have the following typical chemical composition:

$(CH_3)_3SiO[SiO(CH_3)_2]_nSi(CH_3)_3$

Commercial bulk-polymerized dimethyl silicone fluids, such as DOW CORNING 200 fluids, 50-1000 cSt., typically contain trace amounts of process impurities.

INCOMING INSPECTION

Dow Corning recommends that incoming inspection tests be performed to confirm product identity and condition on arrival. Suggested tests include viscosity and infrared identification, and any other tests deemed necessary for the application. Such tests may or may not be run routinely by Dow Corning as lot acceptance tests. Obtain the Sales Specifications for lot acceptance tests and test limits conducted on DOW CORNING 200 fluids, 50-1000 cSt.

SAFE HANDLING INFORMATION

A Materials Safety Data Sheet, as required under existing federal regulations, is available upon request from Dow Corning Corporation, Midland, Michigan 48640.

Note: May cause temporary eye discomfort.

SHELF LIFE

Shelf life is the period of time during which a material may be stored under specified conditions in its original unopened container (except for inspection) while retaining the material's sales specifications. Shelf life starts with the date of shipment (unless otherwise specified), and ends on a given date. Continued storage beyond the designated shelf life does not necessarily mean that the material may not be used. However, after the expiration of the designated shelf life, testing of critical properties and redetermination of suitability for contemplated use of the product is imperative.

Dow Corning certifies that DOW CORNING 200 fluids, 50-1000 cSt., will meet sales specification requirements for a period of 12 months from date of shipment.

Storage temperature: ambient.

PACKAGING

DOW CORNING 200 fluids, 50-1000 cSt., are supplied in 40- and 440-lb (18.1- and 199.6-kg) containers, net weight. Smaller containers are available from repackagers.

Caution! Containers will have product residues when emptied. Follow precautions recommended for handling these products when disposing of the container. Containers are not intended for reuse.

SALES SPECIFICATIONS

Sales specifications information, including detailed test methods and analysis procedures used by Dow Corning, is available upon request. Since Dow Corning reserves the right to update sales specifications information without prior notice, users should periodically request this information.

PRODUCT CHARACTERISTICS

DOW CORNING 200 fluids, 50-1000 cSt., have the following product characteristics:

- Clear
- Essentially Nontoxic
- Nonbioaccumulating
- Nonbioactive
- Nongreasy
- Nonocclusive
- Nonrancidifying
- Nonstinging on Skin
- Tasteless

DOW CORNING 200 fluids, 50-1000 cSt., when compared to other materials that may be substituted in a given application, may offer one or more of these comparative characteristics:

- High Compressibility
- High Dampening Action

TYPICAL PHYSICAL PROPERTIES

Physical properties vary from lot to lot and should not be used for writing sales specifications.
SPECIFICATION WRITERS! Before writing a specification, obtain Dow Corning's Sales Specification on the product.
Contact "Specification Coordinator, Dow Corning, Box 1767, Midland, MI, 48640."

As Supplied	DOW CORNING 200 Fluid, 50 cSt.	DOW CORNING 200 Fluid, 100 cSt.	DOW CORNING 200 Fluid, 200 cSt.	DOW CORNING 200 Fluid, 350 cSt.	DOW CORNING 200 Fluid, 500 cSt.	DOW CORNING 200 Fluid, 1000 cSt.
Appearance	← Crystal clear liquid free from suspended matter and sediment. →					
Specific Gravity @ 25°C	0.960	0.964	0.967	0.968	0.969	0.970
Refractive Index @ 25°C	1.4022	1.4030	1.4032	1.4034	1.4034	1.4035
Color, APHA	5	5	5	5	5	5
Flash Point, open cup, °F	605	>620	>620	>620	>620	>620
Acid Number, BCP	trace	trace	trace	trace	trace	trace
Pour Point, °C	-70	-65	-65	-65	-50	-50
Surface Tension @ 25°C, dynes/cm	20.8	20.9	21.0	21.1	21.1	21.2
Volatile Content, @ 150°C, percent	0.3	0.02	0.07	0.09	0.15	0.11
Viscosity Temperature Coefficient	0.59	0.60	0.60	0.60	0.60	0.61
Coefficient of Expansion, cc/cc/°C	0.00104	0.00096	0.00096	0.00096	0.00096	0.00096
Thermal Conductivity @ 50°C, gm cal/cm · sec · °C	—	0.00037	—	0.00038	—	0.00038
Specific Heat @ 25°C, cal/gm/cm	—	0.352	—	0.350	—	0.349
Solubility Parameter*	7.3	7.4	7.4	7.4	7.4	7.4
Solubility in Typical Solvents,						
Chlorinated solvents	High	High	High	High	High	High
Aromatic solvents	High	High	High	High	High	High
Aliphatic solvents	High	High	High	High	High	High
Dry alcohols	Poor	Poor	Poor	Poor	Poor	Poor
Water	Poor	Poor	Poor	Poor	Poor	Poor
Fluorinated propellents	Poor	Poor	Poor	Poor	Poor	Poor
Dielectric Strength @ 25°C, volts/mil	400	400	400	400	400	400
Volume Resistivity @ 25°C, ohm-cm	1.0×10^{15}	1.0×10^{15}	1.0×10^{15}	1.0×10^{15}	1.0×10^{15}	1.0×10^{15}

*Fedors Method: R.F. Fedors, Polymer Engineering And Science, Feb. 1974.

Dow Corning does not routinely test all these physical properties. Users should independently test these properties when they are critical in the application.

- High Dielectric Strength
- High Oxidation Resistance*
- High Shearability Without Breakdown
- High Spreadability
- High Temperature Serviceability*
- High Water Repellency
- Low Environmental Hazard
- Low Fire Hazard*
- Low Odor
- Low Reactivity*
- Low Surface Energy
- Low Temperature Serviceability
- Low Toxicity
- Low Vapor Pressure
- Good Heat Stability*
- Good Leveling and Easy Rubout
- Soft Feel and Subtle Skin Lubricity

APPLICATION INFORMATION

DOW CORNING 200 fluids, 50-1000 cSt., are not intended for food or medical use. They are intended for use by industrial manufacturers. Typical end uses include:

- Cosmetic Ingredient
- Elastomer and Plastics Lubricant
- Electrical Insulating Fluid
- Foam Preventative or Breaker
- Household Products Ingredient
- Mechanical Fluid
- Mold Release Agent
- Personal Care Products Ingredient
- Polish Ingredient
- Specialty Chemical Products Ingredient
- Specialty Cleaner Ingredient
- Surface Active Agent

*See "Contamination and Fire Prevention."

CONTAMINATION AND FIRE PREVENTION

At elevated temperatures, DOW CORNING 200 fluids, 50-1000 cSt., are sensitive to contamination by strong acids, bases, some metallic compounds and oxidizing agents. These contaminants may cause an accelerated rate of volatile by-product formation. Oxidizing agents can also cause an increase in fluid viscosity. When these conditions may exist, it is recommended that the flash point of the fluids be checked periodically to monitor operational safety. Also, ignitable conditions may exist if the fluid is giving off smoke.

Note: For answers to any questions regarding the efficacy, safety, health or environmental aspects of using DOW CORNING 200 fluids, 50-1000 cSt., in any application, contact your nearest Dow Corning sales office or call the Dow Corning Customer Service. Telephone:

In Michigan (800) 292-2323
Outside Michigan (800) 248-2345

The information and data contained herein are based on information we believe reliable. You should thoroughly test any application, and independently conclude satisfactory performance before commercialization. Suggestions of uses should not be taken as inducements to infringe any particular patent.

DOW CORNING CORPORATION
MIDLAND, MICHIGAN 48640

DOW CORNING

APPENDIX B

Geometry Optimization Program

```
FTN7X,S,L
$FILES (0,3)
$CDS ON
C
C  HIS PROGRAM CALCULATES THE FORCES PRESSURES AND SPRING RATE OF A FOUR
C BAR LINKAGE SUSPENSION UNIT FOR AN AMPHIBIUOS ARMORED PERSONNEL CARRIER
C FOR  DAVID TAYLOR RESEARCH CENTER USING A LIQUID SPRING STRUT
C
C   REVISED 11/5/87 GRW TO INCLUDE A TWO VARIABLE SEARCH
C
      PROGRAM DTOPT
      COMMON RA,FHS,PSJ,HH,K,PP,KIT,KBFAILO,VF,VR,FH2,SRS,DSRM
     +   ,STK,STL,SP1,AAM,Z,BZI
      DIMENSION A(7,7), B(7,7), C(7,7)
      REAL K
C
      OPEN (UNIT = 30,FILE='DTOPTI.DATA',ERR=950)
      OPEN (UNIT = 40,FILE='DTOPTO.DATA',ERR=950)
      OPEN (UNIT = 50,FILE='DTOPTO2.DATA',ERR=950)
C
      READ(30,*)Y1,Y2,X1,RHS,RHL,AZ,RA,FHS,PSJ,HH,K,PP
C     WRITE(40,50)Y1,Y2,X1,RHS,RHL,AZ,RA,FHS,PSJ,HH,K,PP
      WRITE(1,50)Y1,Y2,X1,RHS,RHL,AZ,RA,FHS,PSJ,HH,K,PP
   50 FORMAT(/,'    Y1    Y2    X1   RHS   RHL   AZ    RA     FHS     PSJ    ',
     +'  HH      K       PP',/,
     + 7(1X,F4.1),2(1X,F6.0),1X,F4.1,1X,F7.0,1X,F4.1)
      WRITE(50,51)
   51 FORMAT(/,'    Y1     Y2     X1    RHL     AZ    RHS    AAMX    A4M ',
     + ' VF   VR    FHJ   SRS DSRM STL   STK   SPS     Z     BZI')
   52 FORMAT(4(1X,F5.2),1X,F5.3,3(1X,F5.2),
     + 1X,F5.1,1X,F4.1,1X,F5.0,1X,F3.0,1X,F3.0,
     + 2(1X,F4.1),1X,F4.0,2(1X,F4.1))
C
     WRITE(1,*)'M0'
       AZ = AZ*3.141592/180.     !CONVERT DEGREES TO RADIANS
C     SET UP INPUT DATA ARRAYS
C         1 = CURRENT VARIABLE
C         2 = INITIAL VARIABLE
      B(1,1) = Y1    !Y BOTTOM OF HULL TO ROAD ARM PIVOT (UP +)
      B(2,1) = Y2    !Y ROADARM PIVOT TO STRUT HULL PIVOT
      B(3,1) = X1    !X ROADARM PIVOT TO STRUT HULL PIVOT(FORWARD+)
      B(4,1) = RHL   !ROADARM LENGTH PIVOT TO HUB
      B(5,1) = AZ    !ANGLE ROADARM HUB TO STRUT MOUNT LINE AND RHL(CW+)
      B(6,1) = RHS   !LENGTH BETWEEN HUB AND STRUT MOUNT ON ROADARM
C
     WRITE(1,*)'M1'
      DO M=1,6
        B(M,2) = B(M,1)
      END DO
C         3 = CURRENT DELTA B
      B(1,3) = .25
      B(2,3) = .25
      B(3,3) = .25
      B(4,3) = .25
      B(5,3) = .02
      B(6,3) = .25
C         4 = MIN DELTA B
      B(1,4) = .05
      B(2,4) = .05
      B(3,4) = .05
      B(4,4) = .05
      B(5,4) = .004
      B(6,4) = .05
C         5 = MIN B
      B(1,5) = 8.49
      B(2,5) = 8.35
      B(3,5) = -0.001
      B(4,5) = 15.49
      B(5,5) = 0.1544    !0.0
      B(6,5) = 10.0
C         6 = MAX B
      B(1,6) =  8.50     !18.0
      B(2,6) =  8.36     !18.0
      B(3,6) =  0.01
      B(4,6) = 15.50    !16.0    !17.0      !20.0
      B(5,6) =  0.1545
      B(6,6) =  10.01    !13.0
C         7 = STARTING DELTA B
```

```
C       B(1,7) = .25
C       B(2,7) = .25
C       B(3,7) = .25
C       B(4,7) = .25
C       B(5,7) = .02
C       B(6,7) = .25
C
        B(1,7) = .50
        B(2,7) = .50
        B(3,7) = .50
        B(4,7) = .50
        B(5,7) = .04
        B(6,7) = .50
C     DETERMINE INITIAL  A
       KIT = 0
        CALL DTGEO(AA,B,KBFAIL)
              AAMX = AA
              VFM =VF
              VRM = VR
              FH2M = FH2
              SRSM = SRS
              DSRMM = DSRM
        DO J = 1,6
            A(J,1) = AA
            A(J,2) = AA
            C(J,1) = AA
            C(J,2) = AA
        END DO
C
        IF(KBFAIL.GT.0) THEN
          WRITE(1,*)'INITIAL CONDITIONS OUT OF BOUNDS, DO YOU WISH TO CO
     +NTINUE?    1=YES, 0=NO'
        READ(1,*)KINIT
         IF(KINIT.EQ.1) GOTO 35
        GOTO 990
        ENDIF
C     ITTERATE WITH EACH MAXIMUN ASSENT VARIABLE
   35   JM = 1       ! J VARIABLE WITH MAXIMUM ASSENT, INITIAL SETTING
        JM2= 1       ! J2 VARIABLE WITH MAXIMUM ASSENT, INITIAL SETTING
        BJM = B(JM,1)
        BJMJ2 = B(JM,2)
        STLM = STL
        STKM = STK
        SP1M = SP1
        ZM   = Z
        BZIM = BZI
C
       DO KIT = 1,50    !ITTERATION LOOP
C
       IF(KIT.EQ.50)THEN
        WRITE(40,51)
        WRITE(40,52)(B(L,1),L=1,6),AAMX, A4M,VFM,VRM,FH2M,SRSM,DSRMM
     + ,STLM,STKM,SP1M,ZM,BZIM
       ENDIF
C       WRITE(1,*)'M2  KIT=',KIT
C    START CYCLE THROUGH VARIABLES
        KOUNTL1 = 0
C
CC      DO J = 1,6
        DO J = 1,6
C
          IF(J.EQ.1.AND.KIT.GT.1) THEN
           B(JM,1)= BJM
           B(JM,2)= BJM
           B(JMJ2,1)= BJMJ2
           B(JMJ2,2)= BJMJ2
          ENDIF
C
C
        IF(J.EQ.1)THEN
        WRITE(1,52)(B(L,1),L=1,6),AAMX, A4M,VFM,VRM,FH2M,SRSM,DSRMM
     + ,STLM,STKM,SP1M,ZM,BZIM
        WRITE(50,52)(B(L,1),L=1,6),AAMX, A4M,VFM,VRM,FH2M,SRSM,DSRMM
     + ,STLM,STKM,SP1M,ZM,BZIM
          ENDIF
C
          KOUNTL2 = 0
          KBFAIL = 0
        IF(J.EQ.1) A4M = 0         !A(JM,4) VALUE OF MAXIMUM ASSENT, RESET
C
```

```
C  BEGIN EVALUATION OF EACH VARIABLE EFFECT
 490   B(J,1) = B(J,2) + B(J,3)
                 IF(B(J,1).LT.B(J,5)) B(J,1) = B(J,5)
                 IF(B(J,1).GT.B(J,6)) B(J,1) = B(J,6)
C
C      IF(KOUNTL2.GT.6) GOTO 520
       IF(KOUNTL2.GT.10) GOTO 520
        JM2 = J
C
C    BEGIN SECOND LEVEL OFVARIABLE SEARCH
CC     DO J2 = J,6
       DO J2 = J,6
C
CC     IF(J.EQ.1)THEN
CC     WRITE(1,52)(B(L,1),L=1,6),AAMX, A4M,VFM,VRM,FH2M,SRSM,DSRMM
CC     WRITE(50,52)(B(L,1),L=1,6),AAMX, A4M,VFM,VRM,FH2M,SRSM,DSRMM
CC       ENDIF
C
         KOUNTL3 = 0
         KBFAIL = 0
       IF(J2.EQ.J) THEN
C                 A4M2 = A4M       !A(JM,4) VALUE OF MAXIMUM ASSENT, RESET
                  A4M2 = 0         !A(JM,4) VALUE OF MAXIMUM ASSENT, RESET
CC                KOUNTL3 = 6
                  KOUNTL3 = 10
                  GOTO 680         ! SKIP SECOND VARIABLE SEARCH
       ENDIF
C
C  BEGIN EVALUATION OF EACH VARIABLE EFFECT
 690   B(J2,1) = B(J2,2) + B(J2,3)
                 IF(B(J2,1).LT.B(J2,5)) B(J2,1) = B(J2,5)
                 IF(B(J2,1).GT.B(J2,6)) B(J2,1) = B(J2,6)
C
 680   CONTINUE
CC     IF(KOUNTL3.GT.6) GOTO 620
       IF(KOUNTL3.GT.10) GOTO 620
C
 500   CALL DTGEO(AA,B,KBFAIL)
C
C
            C(J2,1) = AA
C
C
CC     IF(J.LE.1) THEN
CC     WRITE(1,*)' J    B(J,1)   A(J,1)       B(J,3)   KBFAIL  KOUNTL2 '
CCC    WRITE(40,*)' J    B(J,1)   A(J,1)       B(J,3)   KBFAIL  KOUNTL2 '
CC     ENDIF
CCCC   WRITE(1,540) J,J2, B(J2,1), C(J2,1), C(J2,3), KBFAIL,KOUNTL3
CCC    WRITE(40,540) J, B(J,1), A(J,1), B(J,3), KBFAIL,KOUNTL2
C
C  CHECK ASSENT
      IF(J2.EQ.J)THEN
        IF(J2.EQ.5) THEN
         C(J2,4) = (C(J2,1)-C(J2,2))/ABS(B(J2,3)*B(4,2))
        ELSE
         C(J2,4) = (C(J2,1)-C(J2,2))/ABS(B(J2,3))
        ENDIF
      ELSE
        IF(J2.EQ.5) THEN
         C(J2,4) = (C(J2,1)-C(J2,2))/(ABS(B(J2,3)*B(4,2))+ABS(B(J,3)))
        ELSE
          IF(J.EQ.5)THEN
           C(J2,4) = (C(J2,1)-C(J2,2))/(ABS(B(J,3)*B(4,2))+ABS(B(J2,3)))
          ELSE
           C(J2,4) = (C(J2,1)-C(J2,2))/(ABS(B(J2,3))+ABS(B(J,3)))
          ENDIF
        ENDIF
      ENDIF
C
CCC   IF(AAM.LT.0.999.AND.C(J2,4).GE.0.) !GEOMETRY OUT OF BOUNDARIES
CCC  +   C(J2,4) = -C(J2,4)       ! REVERSE ASSENT TO SIGNAL WRONG WAY
CCC    ENDIF
C
CCC    WRITE(1,540) J,J2, B(J2,1), C(J2,1), C(J2,4), KBFAIL,KOUNTL3
      IF(C(J2,4).GE.0.) THEN
C
           IF(C(J2,4).GT.A4M2) THEN       !NEW MAXIMUM ASSENT VALUE
             A4M2 = C(J2,4)
             AAMX2 = AA
             JM2 = J2
```

```
            BJM2 = B(J2,1)
            VFM2 =VF
            VRM2 = VR
            FH2M2 = FH2
            SRSM2 = SRS
            DSRMM2 = DSRM
            STL2 =STL
            STK2 = STK
            SP12 = SP1
            Z2 = Z
            BZI2 = BZI
          ENDIF
C
      ELSE
C         VARIABLE GOING IN WRONG DIRECTION
          IF(KOUNTL3.EQ.0) THEN
            B(J2,3) = -B(J2,3)
          ELSE
            B(J2,3) = -B(J2,3)*.70
          ENDIF
            KOUNTL3 = KOUNTL3 + 1
C
          GO TO 690
      ENDIF
C
C540   FORMAT(I3,2X,F8.3,2X,E9.5,2X,F8.3,2X,I8,5X,I3)
C
  620 CONTINUE
C      RESET B
          IF(B(J2,1).GE.B(J2,2)) THEN     !RESET DELTA B TO RIGHT SIGN
             B(J2,3) = B(J2,7)
           ELSE
             B(J2,3) = -B(J2,7)
          ENDIF
          B(J2,1) = B(J2,2)
C
C 650 CONTINUE
       END DO      ! END VARIABLE LOOP
C   END LOOP OF SECOND LEVEL
C
           A(J,1) = AA
           A(J,1) = AAMX2
           A(J,4) = A4M2
C
C
      IF(J.LE.1) THEN
CCCC  WRITE(1,*)' J  J2  B(J,1)  A(J,1)     B(J,3)  KBFAIL  KOUNTL2 '
CCC   WRITE(40,*)' J  J2  B(J,1)  A(J,1)     B(J,3)  KBFAIL  KOUNTL2 '
      ENDIF
CCCCC WRITE(1,540) J, JM2,B(J,1), A(J,1), B(J,3), KBFAIL,KOUNTL2
CCC   WRITE(40,540) J, JM2, B(J,1), A(J,1), B(J,3), KBFAIL,KOUNTL2
C
C  CHECK ASSENT
CC     IF(J.EQ.5) THEN
CC     A(J,4) = (A(J,1)-A(J,2))/ABS(B(J,3)/B(J,4))
CC     ELSE
CC     A(J,4) = (A(J,1)-A(J,2))/ABS(B(J,3))
CC     ENDIF
       IF(A(J,4).GE.0.) THEN
C
          IF(A4M2.GT.A4M) THEN      !NEW MAXIMUM ASSENT VALUE
            A4M = A4M2
            AAMX = AAMX2
            JM = J
            JMJ2 = JM2
            BJM = B(J,1)
            BJMJ2 = BJM2
            VFM =VFM2
            VRM = VRM2
            FH2M = FH2M2
            SRSM = SRSM2
            DSRMM = DSRMM2
            STLM = STL2
            STKM = STK2
            SP1M = SP12
            ZM = Z2
            BZIM = BZI2
           GOTO 520
C
          ENDIF
```

```
C
      ENDIF
CC    ELSE
C         VARIABLE GOING IN WRONG DIRECTION
          IF(KOUNTL2.EQ.0) THEN
            B(J,3) = -B(J,3)
          ELSE
            B(J,3) = -B(J,3)*.70
          ENDIF
            KOUNTL2 = KOUNTL2 + 1
C
          GO TO 490
CCC   ENDIF
C
 540  FORMAT(2(I3,2X),F8.3,2X,E9.5,2X,F8.3,2X,I8,5X,I3)
C
 520 CONTINUE
C    RESET B
          IF(B(J,1).GE.B(J,2)) THEN
             B(J,3) = B(J,7)
           ELSE
             B(J,3) = -B(J,7)
          ENDIF
          B(J,1) = B(J,2)
C
C 550 CONTINUE
     END DO      ! END VARIABLE LOOP
C
C  RESET ALL A VALUES
        DO J = 1,6
          A(J,1) = A(JM,1)
          A(J,2) = A(JM,1)
          A(J,4) = 0.
          C(J,1) = A(JM,1)
          C(J,2) = A(JM,1)
          C(J,4) = 0.
        END DO
C
      IF(A4M.EQ.0.) GOTO 900
C
C
      END DO      ! END ITTERATION LOOP
C
 900 CONTINUE
C
      WRITE(40,51)
      WRITE(40,52) (B(L,1),L=1,6),AAMX, A4M,VFM,VRM,FH2M,SRSM,DSRMM
    + ,STLM,STKM,SP1M,ZM,BZIM
      KIT = 51
      CALL DTGEO(AA,B,KBFAIL)  !FINAL EVALUATION FOR PLOTTING CURVES
C
      GOTO 990
 950  WRITE(1,*)'CANNOT OPEN INPUT FILE 30'
C
 990  CONTINUE
     CLOSE (UNIT = 30)
     CLOSE (UNIT = 40)
     CLOSE (UNIT = 50)
     STOP
     END
C
C
     SUBROUTINE DTGEO(AA,B,KBFAIL)
C
      DIMENSION SL(48),FS(48),PS(48),FH(48),Y(48),FHF(48),SR(48),
    +           DSR(48),DDSR(48),B(7,7),B1D(48)
      COMMON RA,FHS,PSJ,HH,K,PP,KIT,KBFAILO,VF,VR,FH2,SRS,DSRM
    +   ,STK,STL,SP1,AAM,Z,BZI
      REAL K
C
C     WRITE(1,*) 'S1 ENTER DTGEO'
C
      Y1  = B(1,1)
      Y2  = B(2,1)
      X1  = B(3,1)
      RHL = B(4,1)
      AZ  = B(5,1)
      RHS = B(6,1)
C
C     WRITE(1,*)'RA,FHS,PSJ,HH,K,PP'
```

```
C       WRITE(1,*) RA,FHS,PSJ,HH,K,PP
C       WRITE(1,*)'Y1,Y2,X1,RHL,AZ,RHS'
C       WRITE(1,*)Y1,Y2,X1,RHL,AZ,RHS
C       INITIAL CALCULATIONS
        Z = SQRT(Y2**2 + X1**2)
         BZ = ACOS(X1/Z)
         IF((RHL*SIN(AZ)).GE.RHS) THEN
C           WRITE(1,*)'BAD DATA RHL=',RHL,' * SIN(AZ):AZ=',AZ,' GE RHS=',
C     +              RHS
         AA = 0.05   ! ASSIGN A SMALL FIGUTE OF MERIT
         GOTO 100
         ENDIF
C
         RSL = RHL*COS(AZ) + RHS*COS(ASIN((RHL*SIN(AZ))/RHS))
C      CHECK BOUNDARIES
         KBFAILO = KBFAIL
C        IF(Z.LE.8.35) THEN
         IF(Z.LE.7.35) THEN
                       KBFAIL = KBFAIL + 1
C             AA1 = Z/8.35
              AA1 = Z/7.35
CCCC           WRITE(1,*)'Z=',Z,' LT 8.35',' AA1= ',AA1
CCC            WRITE(40,*)'Z=',Z,' LT 8.35',' AA1 =',AA1
         ELSE
          AA1 = 1.
         ENDIF
C
         Y(1)= 15.5      !16       !STATIC
         Y(2)= -1.5      !-1       !FULL JOUNCE
         Y(3)= 19.5      !20       !FULL REBOUND
         Y(4)= 15.75     !16.25    !STATIC +.25 FOR RATE CALCULATION
         DO I=5,47       !SET Y FOR CYCLING THROUGH FULL TRAVEL
          IF(I.EQ.5) THEN
            Y(I) = Y(3)
          ELSE
            Y(I) = Y(I-1) - 0.5
          ENDIF
         END DO
C
C     DETERMINE POSITION, FORCE, PRESSURE RELATIONSHIPS
       DO I = 1,47
       HP = Y(I) + Y1 - HH
       IF(RHL.LE.0.)WRITE(1,*)'BAD VALUE FOR RHL'
       IF(HP.GE.RHL) THEN
C        WRITE(1,*)'BAD DATA HP=',HP,'--GT RHL=',RHL,'  I=',I,' Y(I)=',
C      + Y(I),' HH =',HH
        AA = .1            ! ASSIGN A SMALL FIGURE OF MERIT
        GOTO 100
        ENDIF
       A1 = ASIN(HP/RHL)
       XPP = X1 + RSL * COS(A1-AZ)
       YPP = Y2 + RSL * SIN(A1-AZ)
       B1 =  ATAN(YPP/XPP)
C      B1D(I) = B1 *   180. /3.141592
       B1D(I) = -B1 * 180./3.141592
         IF(I.EQ.3) THEN
         BZI = Z* SIN(BZ-B1)
         IF(BZI.LT.3.9) THEN
                       KBFAIL = KBFAIL + 10
            AA2 = BZI/3.9
CCCC        WRITE(1,*)'BZI =',BZI,' LT 4.0','B1=',B1,'BZ=',BZ,'Z=',Z
CCCC +  ,'  XPP=',XPP, 'YPP =',YPP, ' Y =',Y(I),' I =',I,' AA2=',AA2
CCCC +  , ' AZ =',AZ, ' A1 =',A1,' RSL=',RSL
CCC         WRITE(40,*)'BZI =',BZI,' LT 4.0','B1=',B1,'BZ=',BZ,'Z=',Z
CCC     +  ,'  XPP=',XPP, 'YPP =',YPP, ' Y =',Y(I),' I =',I,' AA2=',AA2
         ELSE
            AA2 = 1.
         ENDIF
         ENDIF
       SL(I) = XPP/ COS(B1)
       FHF(I) = RSL*SIN(B1-(A1-AZ))/(RHL*COS(A1))
       END DO
C     CALCULATE FORCES PRESSURES AND VOLUMES
       CONTINUE
       FS(1) = FHS/FHF(1)
       PS(1) = FS(1)/RA                ! STATIC
       SP1 = PS(1)
       PS(2) = PSJ                     ! FULL JOUNCE
C     CALCULATE FREE VOLUME REQUIRED TO PROVIDE STATIC AND JOUNCE FORCES
       VF = (SL(1)-SL(2))*RA*K**(1/PP) / (PS(2)**(1/PP) - PS(1)**(1/PP))
```

```
C    CALCULATE PHYSICAL VOLUME AT STATIC
       VS = VF * (1.-(PS(1)/K)**(1/1.3))
C
       DSRM =1.E10     ! RESET MINIMUM WHEEL RATE-RATE
       DDSRM =1.E10     ! RESET MINIMUM WHEEL RATE-RATE
       DO I = 1,47
          V      = VS -(SL(1)- SL(I))*RA    !PHYSICAL VOLUME
          IF(V.GE.VF) THEN
           PS(I) = 0.
           GOTO 60
          ENDIF
C         WRITE(1,*)'I=',I,' V=',V
          PS(I) = K *(1.- (V/ VF))**PP
  60      FS(I) = PS(I) * RA
          FH(I) = FS(I) * FHF(I)
          IF(I.GT.5) THEN        ! CALCULATE EFFECTIVE SPRING RATE
            SR(I) = (FH(I)-FH(I-1))/0.5
            IF(I.GT.6) THEN     ! CALCULATE THE CHANCE IN SPRING RATE
             DSR(I) = (SR(I)-SR(I-1))/0.5
             IF(DSR(I).LT.DSRM) DSRM = DSR(I)   !REMEMBER MINIMUM DSR
              IF(I.GT.7) THEN
               DDSR(I) = (DSR(I)-DSR(I-1))/0.5
               IF(DDSR(I).LT.DDSRM) DDSRM = DDSR(I)
              ENDIF
            ENDIF
          ENDIF


           IF(KIT.EQ.0)
     +     WRITE(1,*)'I=',I,'SL=',SL(I),'PS= ',PS(I),' FS=',FS(I),
     +     ' FH=',FH(I)
           IF(KIT.EQ.51.AND.I.EQ.5)
     +                WRITE(40,38)
  38     FORMAT('   Y   F(HUB)      F(STRUT)   P(STRUT)   SPRING RATE   STRUT',
     +' ANGLE')
           IF(KIT.EQ.51.AND.I.GE.5)
     +                WRITE(40,39)Y(I), FH(I),FS(I),PS(I),SR(I),B1D(I)
  39     FORMAT(1X,F4.1,1X,F8.1,4(2X,F8.1))
       END DO
C
       VR  = VF/((SL(3)- SL(2))*RA)
       IF(VR.GE.20) THEN
               AA3 =  20./VR
             IF(KIT.GT.0)  KBFAIL = KBFAIL+100
CCCC                WRITE(1,*)'VR =',VR,' GT 14',' AA3 =',AA3
CCC                 WRITE(40,*)'VR =',VR,' GT 14',' AA3=',AA3
            ELSE
              AA3 = 1.0
         ENDIF
             IF(DSR(I).LT.0.) THEN
                                KBFAIL = KBFAIL + 1000   !SR MUST NOT DROP
                   AA4 = 1.+ DSRM/50.
CCCC                 WRITE(1,*)'DSRM= ',DSRM,' LT.0',' AA4=',AA4
CCC                  WRITE(40,*)'DSRM= ',DSRM,' LT.0',' AA4=',AA4
                   ELSE
                     AA4 = 1.
                  ENDIF
       SRS =-(FH(4)-FH(1))/ 0.25
C
C    STRUT TARE LENGTH
C     STROKE
       STK = SL(3)-SL(2)
       STL = SL(2)-STK
C      AA5 = STL/10.0
       IF(STL.LE.9.6) THEN
                     AA5 = STL/9.6
                     KBFAIL = KBFAIL + 10000
        ELSE
                     AA5 = 1.
        ENDIF
C
        IF(SRS.LT.528.) THEN
           AA6 = 1. + (SRS-528.)/500.
C          AA6 = 1.
        ELSE
C          AA6 =  1.+ (SRS-528.)/5000.
           AA6 =  1.
        ENDIF
C
C
```

```
C     FIGURE OF MERIT
C
C           VOLUME   VOLUME RATIO    JOUNCE FORCE      MIN RATE RISE
C             1           1.0             1                .1
C        AA = 300./VF + 10./VR + 2.0*(1.- ABS((FH(2)/(3.5*FHS))-1.))+
         AA = 300./VF + 10./VR + 2.0*(1.- ABS((FH(2)/(3.5*FHS))-1.))**2 +
C       +                    DDSRM/200.    !/50.
        +    DSRM/15./40.  + DDSRM/50.
         AAM = AA1 * AA2 * AA3 * AA4 *AA5 * AA6
         AA = AA * AAM
         FH2 = FH(2)
100      CONTINUE
C        WRITE(1,*)'VF,VR,FH(2),SRS,DSRM,AA,AAM',VF,VR,FH(2),SRS,DSRM,AA
C       +          ,AAM
C        WRITE(40,*)'VF,VR,FH(2),SRS,DSRM,AA,AAM',VF,VR,FH(2),SRS,DSRM,AA
C       +         ,AAM
C        WRITE(1,*)' AA=',AA,' AAM=',AAM,' AA1-5 = ',AA1,AA2,AA3,AA4,AA5
C
         IF(KBFAIL.LE.KBFAIL0) KBFAIL = 0
C
C 110    FORMAT(/,'        RA          VF          VR        SRS',/,
C       +  4(1X,F9.1),//,
C       +'  HEIGHT     FORCE      PRESSURE  STRUT FORCE')
C 111    FORMAT(4(1X,F9.1))
C
          END
```

Input File DTOPTI.DATA

```
8.00 8.35 00. 10. 15.5  9.00 1.767 5167. 18000. 15 422936. 1.3
```

Output File DTOPTO.DATA

Y1	Y2	X1	RHL	AZ	RHS	AAMX	A4M	VF	VR	FHJ	SRS	DSRM	STL	STK	SPS	Z	BZI
8.50	8.35	0.00	15.49	.154	10.01	3.06	0.00	313.8	16.1	18582	503	21.	9.6	11.1	5872	8.4	4.4

Y	F(HUB)	F(STRUT)	P(STRUT)	SPRING RATE	STRUT ANGLE
19.5	3365.8	6468.8	3660.9	0.0	-58.3
19.0	3562.5	6944.9	3930.3	393.4	-55.8
18.5	3769.3	7422.6	4200.7	413.8	-53.4
18.0	3984.7	7903.5	4472.9	430.8	-51.2
17.5	4207.6	8388.3	4747.2	445.8	-49.1
17.0	4437.5	8877.5	5024.0	459.8	-47.0
16.5	4674.1	9371.6	5303.7	473.2	-45.0
16.0	4917.3	9871.0	5586.3	486.4	-43.1
15.5	5167.0	10375.8	5872.0	499.3	-41.2
15.0	5423.2	10886.3	6160.9	512.4	-39.4
14.5	5686.1	11402.9	6453.3	525.8	-37.6
14.0	5955.6	11925.6	6749.0	539.1	-35.8
13.5	6232.0	12454.4	7048.3	552.7	-34.1
13.0	6515.3	12989.7	7351.3	566.6	-32.4
12.5	6805.7	13531.5	7657.9	580.8	-30.7
12.0	7103.4	14080.1	7968.4	595.5	-29.0
11.5	7408.6	14635.4	8282.6	610.3	-27.3
11.0	7721.4	15197.5	8600.7	625.6	-25.6
10.5	8042.1	15766.7	8922.9	641.4	-23.9
10.0	8370.8	16343.0	9249.0	657.4	-22.2
9.5	8707.8	16926.5	9579.2	674.0	-20.6
9.0	9053.2	17517.4	9913.6	691.0	-18.9
8.5	9407.5	18115.7	10257.2	708.4	-17.2
8.0	9770.6	18721.5	10595.1	726.3	-15.5
7.5	10142.9	19335.1	10942.3	744.7	-13.8
7.0	10524.7	19956.4	11294.0	763.5	-12.1
6.5	10916.1	20585.6	11650.1	782.8	-10.4
6.0	11317.4	21222.8	12010.7	802.6	-8.6
5.5	11728.8	21868.1	12375.9	822.8	-6.8
5.0	12150.5	22521.8	12745.8	843.5	-5.0
4.5	12582.8	23183.7	13120.4	864.5	-3.2
4.0	13025.7	23854.0	13499.7	885.8	-1.4
3.5	13479.4	24532.9	13883.9	907.4	.5
3.0	13944.0	25220.4	14273.0	929.3	2.4
2.5	14419.6	25916.6	14667.0	951.1	4.3
2.0	14906.1	26621.7	15066.1	973.0	6.3
1.5	15403.3	27335.6	15470.1	994.4	8.4
1.0	15911.1	28058.5	15879.2	1015.5	10.4
.5	16428.9	28790.2	16293.3	1035.6	12.6
0.0	16956.0	29530.9	16712.5	1054.3	14.8
-.5	17491.7	30280.6	17136.7	1071.3	17.1
-1.0	18034.4	31039.0	17565.9	1085.4	19.4
-1.5	18582.4	31806.0	18000.0	1096.0	21.9

APPENDIX C

Composite Pressure Vessel Proposal

ABB

ASEA BROWN BOVERI

May 31, 1988

Mr. Glenn R. Wendel
Research Engineer
Southwest Research Institute
6220 Culebra Rd., San Antonio,
Texas 78284

Dear Mr. Wendel:

Thank you for your interest in our composites pressure bottles. ABB developed pressure bottles have been shown to have high reliability and low weight. We are experienced in the design and manufacturing of both overwound aluminum and the elastomeric liner concept. Although elastomeric liners allow for a lower weight, for high pressure, low permeability and low cost, an aluminum liner is a better choice.

We are pleased to propose the following development program for the pressure cylinder meeting your application requirements.
The proposed program includes all the tasks listed in the following Statement of Work:

- Engineering analysis and design of:
  - the metal liner
  - the reinforced composite layer
- Design and fabricate the tooling required
- Manufacture of composite pressure bottles
- Perform burst test of the first 2 prototypes
- Delivery of 2 prototypes to SWI.

Application requirements are as follows:

Total length	: 15 in. Max.
Outside Diameter	: 7.5 in. Max.
Fluid Volume	: 270 cubic in.
Operating Pressure	: 17,000 psi Max.

ABB Composites Inc

Address 72 Fairbanks
Irvine CA 92718

Telephone
(714) 855-3008

Telefax
(714) 855-6356

ABB

ASEA BROWN BOVERI

The proposed design is:

The composite pressure cylinder shown in Figure 1 is designed to meet the following design requirements:

Burst Pressure	:	28,000 psi Min. at 75 deg. F
		20,000 psi Min. at 250 deg. F

ABB Composites Inc. is offering the composite reinforced pressure cylinder at the following price:

1. Prototype program :

a. Perform all tasks as listed in the Statement of Work above: $14,500.

There is a one time tooling charge of $ 2,500, included in the above price.

The total cost for the prototype program including 2 deliverable prototypes is

$ 14,500

b. Deliver 12 preproduction prototypes at $ 1,200 per unit
total $ 14,400

2. Production rough order of magnitude estimated pricing:

Quantity	Unit Price
1000	$ 500

The price quoted above assumes that the whole quantity is ordered and will be delivered in one lot.

The prototype bottles can be delivered within 14 weeks, F.O.B our plant after receipt of order and the final specification. Please call if any questions arise.

Sincerely,

K. R. Berg, Ph.D.
Vice President, Engineering

ABB Composites Inc

Address 12 Fairbanks Irvine CA 92718 Telephone (714) 855-3008 Telefax (714) 855-6356

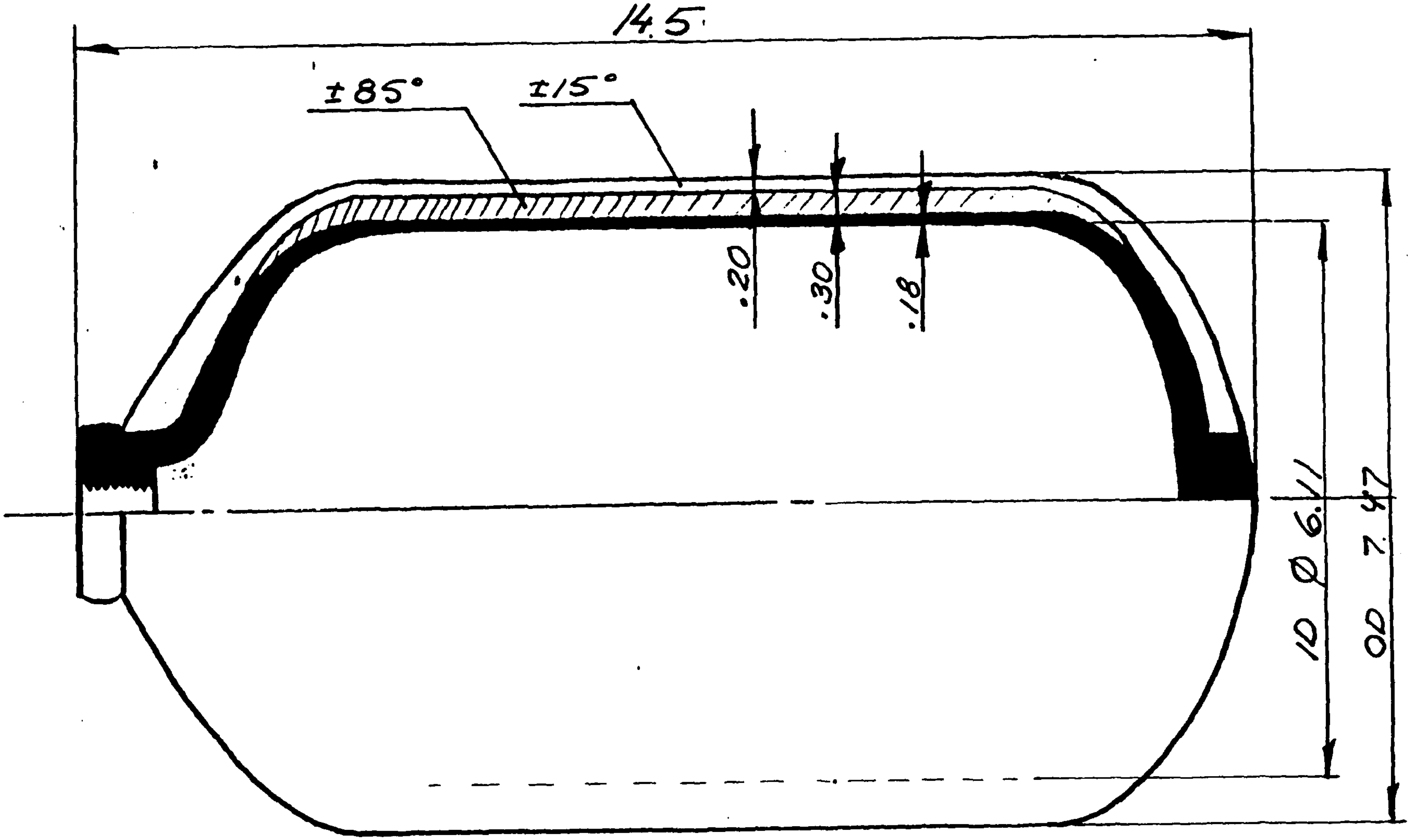

Figure 1: Preliminary pressure bottle